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LOGICAL METHODS OF INVESTIGATION.

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This paper is a series of quotations and connecting remarks taken from notes made by one engineer in his attempt to cut a path for himself through the jungle of scientific and pseudo-scientific literature which surrounds the central clearing where the light is a little brighter and the view a little extended. These notes make no pretense of being an exhaustive résumé of the reading of one man, much less an exhaustive résumé of the literature. They merely indicate a few landmarks and blazes on one of the many trails through the jungle. They may shorten the time for some younger engineer to cut himself a similar path. And they do point out some books which will interest one who is interested in the scientific aspect of our profession.

1. *Introduction.*

Genuine problems, as distinguished from "pseudo-problems", are to be solved, if solved at all, by what is known as the "scientific method". The study and practice of scientific naval engineering certainly offers many genuine problems.

"Scientific method is a collective term denoting the various processes by the aid of which the sciences are built up. In a wide sense, any mode of investigation by which scientific or other impartial and systematic knowledge is acquired is called a scientific method. Such methods are of two principal types—technical and logical. A *technical* or technological *method* is a method of manipulating the phenomena under investigation, measuring them with precision, and determining the conditions under which they occur, so as to be able to observe them in a favorable and fruitful manner. A *logical method* is a method of reasoning about the phenomena investigated, a method of drawing inferences from the conditions under which they occur, so as to interpret them as accurately as possible. . . . These methods of reasoning from the available evidence are not really numerous, and are essentially the same in all sciences. . . . Moreover, these logical methods of science are in a very real sense the soul of the technical methods." (1)

In the report of any scientific investigation it is customary, even necessary, to give some account of the technical methods used. It is rare, however, to state what the writer considers to be the logical methods proper to a scientific investigation. The reason for this omission is, of course, obvious since the same logical methods are applicable to all scientific investigations whatsoever, and it would merely burden a writer's report to repeat over and over again his ideas on the subject. But at least once in his life a man desiring to follow scientific or engineering pursuits should make some record, however brief, of the principal ideals and ideas which guide him and of the principal authors and men who have influenced him in logical methods of science. The very writing of such a record will tend to reduce the number of unconscious assumptions which Whitehead shows are a source of trouble to the scientist and of difficulty to his most constructive and friendly critics.

"It often happens, therefore, that in criticising a learned book of applied mathematics, or a memoir, one's whole trouble is with the first chapter, or even with the first page. For it is there, at the very outset, where the author will probably be found to slip in his assumptions. Further the trouble is not with what the author does say, but with what he does not say. Also it is not with what he knows he has assumed, but with what he has unconsciously assumed. We do not doubt the author's honesty. It is his prespicacity which we are criticising." (2)

For the foregoing reasons the writer presents in these notes a brief summary of the "principal ideals and ideas which guide him and of the principal authors and men who have influenced him in the logical methods of science."

2. *Preliminary Requirements of a Scientist.*

A long list of desiderata might be prepared for the scientist (a term which for brevity will hereinafter be used for the type of engineer to which these notes apply), ranging from various specific mental abilities to ethical habits, but they would apply equally well to the requirements of any other good citizen. In fact, the few requirements listed hereafter apply to any effective man—but they apply with peculiar force to the scientist. He should:

a. Cultivate a passion for intellectual honesty.
b. Speak exactly.
c. Speak clearly.
d. Separate what he knows by observation or trustworthy report from what he infers or learns by unverified report.
e. Limit investigations to genuine problems.
f. Refrain from attempting too wide a field of inquiry.
g. Continue throughout his life the quest for understanding.

The scientist should *cultivate a passion for intellectual honesty*. Scientists are human and as such are subject to the prejudices, warped judgments, etc., arising from human emotions. Now it is difficult to combat an emotion with a logical reason. A better chance for reason is offered when one emotion combats another.

Hence, the recommendation to cultivate *passion* for intellectual honesty, which is the guiding spirit of science.

"And this guiding spirit, this Spirit of Science, is a desire for *truth*, so strong and so compelling that prejudices and preconceptions are burned away, leaving the mind crystal clear to perceive the significance of fresh knowledge and adjust it harmoniously to knowledge already acquired.

"The general method of science is thus simply summarized: 'to explore the facts of nature, to ascertain their mutual relations, and to arrange them as far as possible into a consistent and intelligible whole.'

"What specifically distinguishes modern science from its forerunners is its unswerving quest for objective truth, attained through exact and verifiable knowledge." (3)

The scientist should *speak exactly*. This advice is ancient. About 3550 B.C. Ptah-Hotep advised his son: "If thou have entered as an expert, speak with exact (?) lips, that thy conduct may be seemly." (4) Ptah-Hotep's advice is good today.

The scientist should *speak clearly*. "Some people conceive well and deliver poorly, for without clarity the children of the soul—concepts and judgments—are not born." (5) Or to take the more ancient advice which Paul gave the Corinthians about 56 A.D., "So also ye, unless ye utter by the tongue speech easy to be understood, how shall it be known what is spoken, for ye will be speaking into the air." (6) This does not mean, however, that the scientist should consider himself limited to "short Anglo-Saxon words".

"An elementary word used to describe a complex thing means of necessity inexactness. The edges are blurred. . . . Mature thinking is discriminating thinking. Discriminating thinking demands subtle and exact phrasing. To be concise is not always to be clear. Often it is only to be incomplete and vague." (7)

The scientist should *separate what he knows by observation or trustworthy report from what he infers or learns by unverified report*. He should, in fact, follow the example of Herodotus as reported by Rascoe:

"But there is one peculiarity about Herodotus which should

be observed and that is the careful distinction he always made as to (1) what he himself knew to be true after careful investigation, (2) what he had an opinion about—the reasons for his opinion he is usually careful to set forth,—and, (3) what he heard from others and set down, for what it is worth, without any opinion about it at all." (8)

The scientist should *limit his investigations to genuine problems.* As Johnson (9) says, the scientist asks first: Is there a possible experimental procedure which will yield an answer? i.e., is there a genuine problem. In "genuine problems" experimental procedure can be conceived and executed. In problems involving "absurd requirements" experimental procedure can be conceived but not executed. In "pseudo-problems" an experimental procedure cannot even be conceived. "Experimental procedure" as used above involves its broadest connotations. For example, an astronomer certainly poses himself a genuine problem when he asks: When will the next total eclipse of the sun occur? He cannot manipulate the sun and moon as he does objects in the laboratory—but he can place himself and his instruments in position to take observations by which he can solve his problem. By his choice of instruments for, and position and time of, observation the astronomer has in a broad, and I believe a correct sense, conceived and executed an experimental procedure. In a similar way it is believed that an economist or a social psychologist may be concerned with genuine problems, conceiving and executing experimental procedure; even though, like the astronomer, they cannot manipulate the objects they observe.

The scientist should *not attempt too wide a field of inquiry.* The older I become, the more it seems to me that the most fruitless, the most useless, and often the most pernicious books are those which purport to solve any or all problems with complete finality, whether they be problems of science, religion, philosophy, ethics, or economics. The scientist, at least, should recognize his finite mind and be more modest.

"Any individual, anywhere, who presents some faultless doctrine guaranteed completely and fully to explain any segment of the universe is, by that token, not a scientist. No true scientist

could explain a common pin within his lifetime, though he devoted himself exclusively to that minute object." (10)

"The search for Truth is in one way hard and in another easy. For it is evident that no one can master it fully nor miss it wholly. But each adds a little to our knowledge of Nature, and from all the facts assembled there arises a certain grandeur." (11)

"I may emphasize that both scientific discovery and its practical application are the products of long and arduous research. Discovery and invention do not spring full grown from the brains of men. The labor of a host of men, great laboratories, long, patient scientific experiment, build up the structure of knowledge, not stone by stone, but particle by particle. This adding of fact to fact some day brings forth a revolutionary discovery, an illuminating hypothesis, a great generalization, or practical invention." (12)

"No theory can be expected to prophesy correctly forever, or to anticipate all the data that are to be obtained." (13)

"Probably the most certain thing, in scientific experience, is that all judgments, however they may originate, will be modified in the future." (14)

"Science, in fact, deals only with limiting cases and has to discover and define the range within which its deductions hold good, and when these limitations are borne in mind there is probably no theoretical result, however strange or bizarre it may appear, which will not ultimately prove applicable in some special field." (15)

The scientist should *continue throughout his life the quest for understanding*. The last words in the last book of Alfred the Great express the idea: "Therefore he seems to me a very foolish man, and a very wretched, who will not increase his understanding while in this world, and ever wish and long to reach that endless life where all shall be made clear." (16)

3. *The Distinguishing Characteristics of Scientific Activities.*

Several characteristics distinguish scientific from nonscientific activities. Among them are:

1. The adoption of a special axiom—the "*Uniformity of Nature*".

2. The criteria of the validity of knowledge, for example, science:

a. Accepts the objects of immediate perception, i.e., raw sense data.

b. Accepts a series of universals and relations–formal truths not proved by observations. A universal truth is one for which no contradiction, or exception, can be found or imagined.

c. Accepts all the axioms of Aristotelian Logic, or "Laws of Thought". For example: Law of Identity–All A is A; Law of Contradiction–No A is Not A; Law of the Excluded Middle–Every member of the universe is A or not A; the "Canon of Aristotle"–Part of a part is part of a whole; and all the propositions which can be adduced from and by the above.

d. Accepts all the axioms of conventional algebra.

e. Accepts all the axioms of some geometry, which until recently was Euclid's geometry exclusively. Lately other geometries have been used to give a valid and sometimes more convenient description. (Note: In the study of naval engineering, Euclid's geometry is most convenient and should generally be used.)

The foregoing comments are largely based on Johnson's lectures.(17)

It is desired to interpolate at this point the statement that Heisenberg's "Principle of Uncertainty (or Indeterminacy)" (18) (19) in no way runs counter to or contradicts the principle of the "Uniformity of Nature".

The "Principle of Uncertainty (or Indeterminacy)" reduces to this: We cannot "determine" accurately both the position *and* the velocity of an electron at the same time. J. E. Turner (20) has pointed out that confusion has arisen between two meanings of the verb "to determine"–namely "to ascertain" and "to cause".

The correct meaning in the "Principle of Uncertainty (or Indeterminacy)" is "to ascertain". Hence the principle merely indicates that there is a finite limit to our precision of measurement. It does not in any way run counter to physical determinism or the "Uniformity of Nature".

The scientist is concerned with making correct judgments.

Collier (21) shows that judgment is a mental act in which an affirmation or negation is made with a conviction of its validity for the world of fact or of reason. Deductive reason proceeds from the general to the particular. Inductive reasoning proceeds from particulars to the general. Analogical reasoning proceeds from particulars to particulars. Intuition is *immediately* grasping a particular or a universal truth (*Cf.* Johnson's accepting "objects of immediate perception" and "a series of universals and relations"). Demonstration cannot begin with demonstration—therefore it must begin with intuitive thought—with an ultimate reality which must be felt.

"In fact nothing which is really ultimate can ever be defined or described, but can merely be 'pointed out' by describing complex situations in which it occurs." (22)

4. *The Law of Parsimony.*

Essentially this law says that we shall not multiply entities, hypotheses, explanations, etc., needlessly. This law has appeared again and again among the principles of the greatest thinkers. John Duns Scotus (c. 1265-1308) was one of the earliest to object to the multiplication of entities.(23) "Occam's Razor" (William of Occam, c. 1300-1349), is also an objection to the multiplication of entities—"*Entia non sunt multiplicanda praeter necessitatem*" (We must not assume the existence of any entity unless we are compelled to do so.)

Sir Isaac Newton (24) makes this Rule I in his "Rules of Reasoning", which are worth quoting in full:

"Rule I.—We are to admit no more causes of natural things than such as are both true and sufficient to explain their appearances."

"Rule II.—Therefore, to the same natural effects we must, as far as possible, assign the same causes."

"Rule III.—The qualities of bodies, which admit neither of intension nor remission of degrees, and which are found to belong to all bodies within the reach of our experiments, are to be esteemed the universal qualities of all bodies whatsoever."

"Rule IV.—In experimental philosophy we are to look upon propositions collected by general induction from phaenomena

as accurately or very nearly true, notwithstanding any contrary hypothesis that may be imagined, till such time as other phaenomena occur, by which they may either be made more accurate, or liable to exceptions."

Pearson (25) goes still further. He considers that the fundamental canon of the scientific method is:

"To no concept, however invaluable it may be as a means of describing the routine of perceptions, ought phenomenal existence to be ascribed until its perceptual equivalent has been actually disclosed."

"Whenever we disregard this canon, when, for example, we assert reality for the mechanisms by aid of which we describe our physical experience, then we are more likely than not to conclude with an antimony, or a conflict of rules." (25)

5. Deductive and Inductive Methods.

De Juhasz writing on methods used to investigate spray characteristics named very concisely the four major procedures used in scientific investigations.

"By earlier investigators three methods have been considered for the investigation of spray characteristics: The empirical, the causal and the dimensional method."

"The *empirical method* aims to determine the dependence of spray characteristics on the basis of experimental facts. The *causal method* endeavors to derive the functional relationship by means of theoretical deductions from basic natural laws. The *dimensional method* is a combination of the former two."

"There is another avenue of approach through which some properties of the spray are accessible: the *statistical method* which will be discussed in the following."

"This method is based on the theory of probability which establishes a relationship between the magnitude of aberations from the mean of a large number of similarly influenced phenomena, and their respective frequency of occurrence." (26)

The writer has had to make considerable use of both statistical and dimensional methods in many investigations. For guidance in the use of the statistical method he has relied upon the works of Yule, Jerome, Kelly and Mills. (27) (28) (29) (30)

Other statistical books which are very useful are those by Croxton and Cowden, Snedecor, and Shewart. (31) (32) (33)

For guidance in the use of dimensional methods he has relied largely upon the works of Buckingham, Bridgman, and Porter. (34) (35) (36) (37) (38) The most outstanding example I have seen of a naval officer acquiring mastery of dimensional analysis and then applying it is presented in the works of Conrad.(39) (40)

The scientific idea of "cause" and of "law" must be clearly understood before we can use the causal method effectively. Pearson (41) says:

"Scientifically, cause, as originating or enforcing a particular sequence of perceptions, is meaningless—we have no experience of anything which originates or enforces something else. Cause, however, used to mark a stage in a routine, is a clear and valuable conception, which throws the idea of cause entirely into the field of sense-impressions, into the sphere where we can reason and reach knowledge. Cause, in this sense, is a stage in a routine of experience, and not one in a routine of inherent necessity. The distinction is, perhaps, a difficult one, but it is all the more needful that the reader should fully grasp it."

"There is nothing in any scientific cause which compels us of inherent necessity to predict the effect. The effect is associated with the cause simply as a result of past direct or indirect experience."

"We are now in a position to scientifically define *cause*. Whenever a sequence of perceptions D, E, F, G, is invariably preceded by the perception C, or the perceptions C, D, E, F, G always occur in this order, that is, form a routine of experience, C is said to be a *cause* of D, E, F, G, which are then described as its effects. No phenomenon or stage in a sequence has only one cause, all antecedent stages are successive causes, and, as science has no reason to infer a first cause, the succession of causes can be carried back to the limit of existing knowledge, and beyond that *ad infinitum* in the field of conceivable knowledge. When we scientifically state causes we are really describing the successive stages of a routine of experience. Causation, says John Stuart Mill, is uniform antecedence, and

this definition is perfectly in accord with the scientific concept."

"*All knowledge is concise description, all cause is routine.*"

"As to the scientific method, we saw in our first chapter that it consists in the careful and often laborious classification of facts, in the comparison of their relationships and sequences, and finally in the discovery by aid of the disciplined imagination of a brief statement or formula, which in a few words resumes the whole range of facts. Such a formula, we have seen, is termed a *scientific law*. The object served by the discovery of such laws is the economy of thought; the suitable association of conceptions drawn from stored and correlated sense-impresses, permits the fitting exertion to follow with the minimum of thought upon the receipt of an immediate sense-impression. The knowledge of scientific law enables us to replace or supplement mechanical association, or instinct, by mental association, or thought. It is the *forethought*, by aid of which man in a far higher degree than other animals is able to make the fitting exertion on the receipt of a novel group of sense-impressions."

"A natural law is thus seen to be a résumé in mental shorthand, which replaces for us a lengthy description of the sequences among our sense-impressions. Law in the scientific sense is thus essentially a product of the human mind and has no meaning apart from man. It owes its existence to the creative power of his intellect. There is more meaning in the statement that man gives laws to Nature than in the converse that Nature gives laws to man."

"The law of gravitation is a brief description of *how* every particle of matter in the universe is altering its motion with reference to every other particle. It does not tell us *why* particles thus move; it does not tell us *why* the earth describes a certain curve round the sun. It simply resumes, in a few brief words, the relationships observed between a vast range of phenomena. It economizes thought by stating in mental shorthand that routine of our perceptions which forms for us the universe of gravitating matter."

"We have in the law of gravitation an excellent example of a scientific law. . . ."

"How idle is it, then, to speak of the law of gravitation or indeed of any scientific law, as *ruling* nature. Such laws

simply *describe*, they never *explain* the routine of our perceptions, the sense-impressions we project into an 'outside world'."

Dewey (42) adds the following:

"In technical statement, laws on the new basis are *formulae for the prediction of the probability of an observable occurrence.* They are designations of relations sufficiently stable to allow of the occurrence of forecasts of individualized situations—for every observed phenomenon is individual—within limits of specified probability, not a probability of error, but of probability of actual occurrence. Laws are inherently conceptual in character, as is shown in the fact that either position or velocity may be fixed at will. To call them conceptual is not to say that they are merely 'mental' and arbitrary. It is to say that they are *relations* which are thought, not observed. The subject-matter of the conceptions which constitute laws is not arbitrary, for it is determined by the interactions of what exists. But determination of them is very different from that denoted by conformity to fixed properties of unchanging substances. Any instrument which is to operate effectively in existence must take account of what exists, from a fountain pen to a self-binding reaper, a locomotive or an airplane. But 'taking account of', paying heed to, is something quite different from literal conformity to what is already in being. It is an adaptation of what previously existed to accomplishment of a purpose."

As engineers we normally think of *efficient* causation and are likely to write differential equations to give clear expression to our thoughts. But Johnson (43) shows clearly that in addition to *efficient* causation, where temporal sequences occur as described in some of the previous quotations, we may properly speak of *mnemic* causation and *telic* causation. He goes further and shows that in Symbolic Logic we have ready at hand a tool, generally more powerful than our every day mathematics, for handling problems in all three types of causation. His brief paper is one of the kind that may take decades to affect our every day engineering thinking, not because it is obscure—it is very clear and concise—but because it is so deeply fundamental. However, before reading the paper one should acquire at least an elementary knowledge of Symbolic Logic and what it can do.

The most lucid elementary book on this subject I have studied is that by Langer.(44)

Now if we have some well-established "laws" we may use them as premises and deduce the "cause" of certain other phenomena by the deductive method, even going so far as to use the formal syllogism if we desire. Such procedure is, however scientific, more like the procedure of a civil engineer building a bridge. From well-established principles he calculates, i.e., deduces, stresses, strains, etc., and finally designs and builds the bridge. An equivalent bridge may never have been built before, but if it is merely deduced or calculated from existing principles no addition to knowledge has been made.

Deductive methods have, however, an even more important use in science than the application of known principles, as is shown by McClure and Trotter.

"It is the essence of deduction to utilize all that one knows in order to find out what one does not know." (45)

"The writer would contend that while that method (deductive) is admittedly dangerous when used as a substitute for a kind of investigation in which deductive processes are reduced to a minimum, yet it has its special field of usefulness in cases where the significance of previously accumulated facts has been misinterpreted, or where the exacter methods have proved unavailing through the investigator having been without indications of precisely what facts were likely to be the most fruitful subject for measurement."

"In the first place it must be stated with emphasis that deductive speculation of this sort finds its principal value in opening up new possibilities for the application of a more exact method. Science is measurement, but the deductive method may indicate those things which can be most profitably measured." (46)

Such use of deductive methods leads us to discover new facts, to formulate new hypotheses, to perform new experiments, and to reason about them inductively. The whole subject of experiment and the particular inductive reasons applied to it is so important that it will be treated in the following separate subsection.

6. Experimental Methods and the Use of Induction.

Smith in differentiating experimental from statistical methods said:

"The *sine qua non* of the experimental method is control of conditions. The 'five canons of experimentation' boil down to just this: Given control of all conditions, you hold all constant but one, you vary that, and measure what happens in its presence or absence, or with its several degrees of variation. The essence of the experimental method is that it elucidates quantitative data affected by *one* cause. . . . The essence of the statistical mthod is that it elucidates quantitative data affected by a *multiplicity* of causes." (47)

That is akin to Dewey's statements:

"*The* method of physical inquiry is to introduce some change in order to see what other change ensues; the correlation between these changes, when measured by a series of operations, constitutes the definite and desired object of knowledge. There are two degrees of control of change which differ practically but are alike in principle. In astronomy, for example, we cannot introduce variation into remote heavenly bodies. But we can deliberately alter the conditions under which we observe them, which is the same thing in principle of logical procedure. By special instruments, the use of lens and prism, by telescopes, spectroscopes, interferometers, etc., we modify observed data. Observations are taken from widely different points in space and at successive times. By such means interconvariations are observed. In physical and chemical matters closer at hand and capable of more direct manipulation, changes introduced affect the things under inquiry. Appliances and re-agents for bringing about variations in the things studied are employed. *The progress of inquiry is identical with advance in the invention and construction of physical instrumentalities for producing, registering and measuring changes.*"

"From the experimental point of view, the art of knowing demands skill in selecting appropriate sense-data on the one side and connecting principles, or conceptual theories, on the other. It requires a developed and constantly progressive tech-

nique to settle upon both the observational data and the ideas that assist inquiry in reaching a conclusion in any particular case. (But in Kant's view, the distinction and the connection between the two, while necessary to anything which may be termed cognition, have nothing to do with the validity of any particular enterprise of knowing. Illusion and error exemplify the synthesis of sense and understanding quite as much as does the soundest instance of scientific discovery.) In one case, the heart of the whole matter is the exercise of a *differential* control which makes the difference between good and bad knowing. In Kant's scheme the blessings of the categories descend upon the material of sense without reference to making a distinction between the true and the false."

"We summarize the difference as follows. 1. In experimental knowing, the antecedent is always the subject-matter of some experience which has its origin in natural causes, but which, not having been controlled in its occurrence, is uncertain and problematic. Original objects of experience are produced by the natural interactions of organism and environment, and in themselves are neither sensible, conceptional nor a mixture of the two. They are precisely the qualitative material of all our ordinary untested experiences. 2. The distinction between sense-data and interpretive ideas is deliberately instituted by the process of inquiry, for the sake of carrying it forward to an adequately tested conclusion, one with a title to acceptance. 3. Hence each term of the distinction is not absolute and fixed, but is contingent and tentative. Each is subject to revision as we find observational data which supply better evidence, and as the growth of science provides better directive hypotheses to draw upon. 4. Hence the material selected to serve as data and as regulative principles constantly check one another; any advance in one brings about a corresponding improvement in the other. The two are constantly working together to effect a rearrangement of the original experience material in the construction of a new object having the properties that make it understood or known." (48)

One of the most important requisites of an experiment is given by Schroedinger.

"The first and indispensable condition that we demand of any process of experiment before it can be admitted into the regular procedure of physical research is that it will invariably reproduce the same results. We do not consider an experiment worthy of scientific consideration or acceptance unless it can fulfill the condition. Now it is from the immense mass of individual results accruing from such reproducible experiments that the whole texture of Physical Science is woven." (49)

The mere performance of a reproducible experiment is not enough. We must use inductive methods to make inferences from the observed facts. Pearson's "Canons of Legitimate Inference" are a fair guide in many cases. They are:

"1. Where it is impossible to apply man's reason, that is to criticize and investigate at all, there it is not only unprofitable, but antisocial to believe."

"Belief is thus to be looked upon as an adjunct to knowledge: as a guide to action where decision is needful, but the probability is not so overwhelming as to amount to knowledge. To believe in a sphere where we cannot reason is antisocial, for it is a matter of common experience that such belief prejudices action in spheres where we can reason."

"2. We may infer what we cannot verify by direct sense-impression only when the inference is from known things to unknown things of the like nature in similar surroundings."

"Thus we may not infer an 'infinite' consciousness outside the physical surroundings of finite consciousness; we may not infer man in the moon, however like in nature to ourselves, because the physical surroundings in the moon are not such as we find man in here, etc., etc."

"3. We may infer the truth of tradition when its contents are of like character and continuous with men's present experience, and when there is reasonable ground for supposing its source to lie in persons knowing the facts and reporting what they knew."

"The tradition that Wellington and Blucher won the battle of Waterloo fulfills the necessary conditions, while the miracle of Karl the Great and the adder fulfills neither condition."

"4. While it is reasonable in the minor actions of life, where

rapidity of decision is important, to infer on slight evidence and believe on small balances of probability, it is opposed to the true interests of society to take as a permanent standard of conduct a belief based on inadequate testimony."

"This canon suggests that the acceptance, as habitual guides to conduct, of beliefs based on insufficient evidence, must lead to the want of a proper sense of the individual's responsibility for the important decisions of life. I have no right to believe at seven o'clock that a cab will be on the stand at eight o'clock, if my catching the train at half-past is of vital importance to others." (50)

The most important formal rules for induction are believed to be Mill's "Canons of Experimental Methods", as follows:

"*1st Canon—Method of Agreement:*—If two or more instances of the phenomenon under investigation have only one circumstance in common, the circumstance in which alone all the instances agree is the cause (or effect) of the given phenomenon."

"*2nd Canon—Method of Difference:*—If an instance in which the phenomenon under investigation occurs, and an instance in which it does not occur, have every circumstance in common save one, that one occurring only in the former; the circumstance in which alone the two instances differ is the effect, or the cause, or an indispensable part of the cause, of the phenomenon."

"*3rd Canon—Joint Method of Agreement and Difference:*—If two or more instances in which the phenomenon occurs have only one circumstance in common, while two or more instances in which it does not occur have nothing in common save the absence of that circumstance, the circumstance in which alone the two sets of instances differ is the effect, or the cause, or an indispensable part of the cause, of the phenomenon."

"*4th Canon—Method of Residues:*—Subduct from any phenomenon such part as is known by previous inductions to be the effect of certain antecedents, and the residue of the phenomenon is the effect of the remaining antecedents."

"*5th Canon—Method of Concomitant Variations:*—Whatever phenomenon varies in any manner whenever another phenomenon varies in some particular manner, is either a cause or

an effect of that phenomenon, or is connected with it through some fact of causation." (51)

In addition to these "Canons of Experimentation" we sometimes can more profitably use what Wolf calls the "Deductive-Inductive Method".

"The progress of science, and of knowledge generally, is frequently facilitated by supplementing the simpler inductive methods by deductive reasoning from knowledge already acquired. . . . The differences in order of sequences (deduction-induction; or, induction-deduction) are of no great importance, and hardly deserve separate names. What is of importance is to note the principal kinds of occasion which call for use of this combined method. They are mainly three in number: (1) When an hypothesis cannot be verified (i.e., tested) directly, but only indirectly; (2) when it is possible to systematize a number of already established inductions, or laws, under more comprehensive laws or theories; (3) when, owing to the difficulties of certain problems, or on account of the lack of sufficient and suitable instances of the phenomena under investigation it is considered desirable either to confirm an inductive result by independent deductive reasoning from the nature of the case in the light of previous knowledge, or to confirm a deductive conclusion by independent inductive investigation." (52)

The scientific procedure may be summed up as follows:

"The general principles controlling the development of science are five: (1) It must be primarily empirical. (2) It must proceed from its empirical starting point by the construction of working hypotheses. (3) It must subject these hypotheses to experimental test. (4) It must furnish a specific type of proof for accepted hypotheses, and (5) It must be scrupulously exact in the use of terms."

". . . Having observed phenomena, we seek to 'explain' them: and explanation consists in the framing of hypotheses in terms of which can be described not only the phenomena observed, but other phenomena as yet inadequately explained . . ."

"The larger and more important hypotheses can seldom be put to experimental test as wholes, but deductions made from

the hypotheses—really minor hypotheses which are necessarily true if the major hypothesis is true—are tested experimentally . . . *Experiment* is in every case the final test of a hypothesis."

". . . There must always be included the exact statement of the conditions under which another trained observer may observe the same phenomena. The hypothesis or deduction is not proved until these conditions are described so exactly that another person who follows these directions literally, and with no requirement laid upon him except that he shall follow the instructions as given, may obtain the results which are claimed. The nature of this scientific requirement of proof is frequently overlooked . . ." (53)

"In arriving at a scientific law there are three main stages: the first consists in observing the significant facts; the second in arriving at a hypothesis, which, if it is true, would account for these facts; the third in deducing from this hypothesis consequences which can be tested by observation." (54)

"There is a distinction between hypotheses generated in that seclusion from observable fact which renders them fantasies, and hypotheses that are projections of the possibilities of facts already in existence and capable of report. There is a difference between the imaginative speculations that recognize no law except their own dialectic consistence, and those which rest on an observable movement of events, and which foresee these events carried to a limit by the force of their own movement. There is a difference between support by argument from arbitrarily assumed premises, and an argument which sets forth the implications of propositions resting upon facts already vitally significant." (55)

"The essence of the scientific method is the discovery of general laws through the study of particular facts. It is thus a synthesis of the Greek and renaissance outlooks. Particular facts are the basis of the whole structure, but they are used for the purpose of induction, and when they have led to general laws inductively obtained, the Greek methods of deduction are applied to infer new particular facts from the laws. This method has had the most amazing success—amazing, because it is as indefensible intellectually as the purely deductive method of the Middle Ages. Hume long ago showed it up. All philosophy

since his day has consisted of sophistical refutations of his arguments; the special skill of the philosopher has consisted in making his refutations so subtle and obscure that their fallacious character was not apparent. Men of science, meanwhile, have simply ignored Hume, and have marched from triumph to triumph. Gradually, however, more especially during the last thirteen years, the best men of science, as a result of technical progress, have been led more and more to a form of skepticism closely analogous to Hume's. Eddington, in expounding the theory of relativity, tends to the view that most so-called scientific laws are human conventions." (56)

"To the scientist, failure is a means of discovering truth only a little less than success. It proves how a thing cannot be done and, in so doing, closes up one of the by-paths down which investigation might turn." (57)

"The test of the acceptability of any hypothesis is not its conventionality or its grotesqueness, its abstract or its simple nature, but what it will do." (58)

7. *The Use of Mathematics.*

It may be admitted that there is a "pure science" of mathematics, which produces results that are often found to be extremely useful, but which in general is regarded by its votaries as merely the manipulation of meaningless symbols according to fixed rules, and which is regarded by them as "true" provided only that the results are consistent with some given set of axioms or postulates, however preposterous these may be.

The physical scientist probably has more respect for, and makes more use of, mathematics than any other type of scientist. But as long as he remains in his capacity as a physical scientist he never forgets that mathematics is merely a tool—probably the keenest tool ever forged by human intellect—but a tool nevertheless to be used or discarded as suits his needs.

"It is said that, during his long membership in the Yale faculty Willard Gibbs made but one speech, and that of the shortest. After a prolonged discussion of the relative merits of language and mathematics as elementary disciplines, he rose to remark, 'Mathematics is a language'. However, such a language, usually acquired late in life, must not be used unnecessarily in place

of our mother tongue if we wish to avoid an appearance of affectation."

"Mathematics offers a wonderful shorthand for the precise formulation of well standardized ideas. On the other hand, the expressions of mathematics are lacking in humor, which is to say that they are no suitable medium for those finer shades of thought which are often necessary in the exposition of ideas which are on the way towards standardization. The formal severity of mathematical treatment has its disadvantages. Indeed in our opinion absolute mathematical rigor is a sort of ignis fatuus, which must not serve as a guide to the scientific investigation, although we do not claim that its pursuit, with proper safeguards, may not offer a very wholesome exercise."

"In this book we have not consciously sacrificed any desirable elements of mathematical rigor. If we have the appearance of doing so, it is because we feel the great need of a visualization of the numerous problems before us, and because this end seems best to be attained by mitigating rather than accentuating the formality of mathematical analysis. It is a dangerous thing to use any kind of mathematical equation unless we keep its meaning before us, and are able to express this meaning without the symbolism which mathematics affords." (59)

Most engineers have advanced through the various branches of calculus and differential equations. However, if they encounter a problem of great complexity when using differential equations in Cartesian coordinates they will usually find the problem simplified and condensed by using vector and tensor analysis, just as we find that a problem which is hopelessly complex when expressed in words becomes simplified and condensed when expressed in algebra.

It is comparatively easy to think of and manipulate such vector analysis terms as *del, div, curl,* etc., until a problem is solved and then expand the result by writing the equations which define these terms. We would be hopelessly bogged down in a jumble of long equations if we had to think of or write the defining equations each time in the process of solving the problem.

The use of mathematical statistical methods has been mentioned previously. Every engineer should have a little knowledge of

this mathematical tool and a few engineers should have a complete mastery of it. We do not all need that mastery but should know enough to recognize how much this tool can do for us and to be able to formulate problems for our colleagues who have specialized in its use.

Dimensional analysis has also been mentioned previously. We should all have a little knowledge of the subject if for no other reason than because it often helps to detect erroneous formulas. But I am firmly convinced that no human being anywhere at any time should be permitted to direct any important experiment involving the physical sciences unless he has a thorough grasp of dimensional analysis.

A few years ago I would have believed that no engineer need be familiar with such subjects as quantum theory, wave mechanics, relativity, etc., but now I am not so sure. The advances in nuclear physics in the last decade alone may spill over into engineering at any time. It would be good insurance to take a few of our very youngest engineers and give them a thorough grounding in the theory and mathematics they would need if widely publicized predictions of atomic power come true.

For those who are interested in pushing mathematics still further in both directions, and as a matter of fact condensing hard subjects and thus making them easier to think about, there is nothing better than "symbolic logic." The classic works of our time are the three volumes written several decades ago by Whitehead and Russell.(60)

These volumes will probably remain the classic for some decades to come. However, I would not recommend that any beginner in this field start with those volumes. He should cut his teeth on something simpler first; for example, the elementary text by Langer previously mentioned.(44) She not only gives a very clear introduction to the subject, but shows that it can be used for a much wider field of thinking than mathematics alone. For example, Langer, using symbolic logic, condenses the whole of Aristotelian logic into ten pages.(61) If anyone just visualizes the pages of Aristotle he will realize the advantage.

Following a study of Langer's book, one which is a bit more advanced and is a good next step is that by Lewis and Langford.(62) A person who had mastered those two books would

be in a position to begin the study of Whitehead and Russell. He would shortly see that, as Whitehead and Russell state, by the use of symbolic logic "the scope of mathematics is enlarged both by the addition of new subjects and by a backward extension into provinces hitherto abandoned to philosophy." It is this backward extension into fundamentals that is apt to cause the beginner the most difficulty. He is likely to think it is silly to discuss such childish subjects. But if he masters those he will find he can go on into realms that would otherwise be hopelessly complex.

Reiterating earlier statements, mathematics furnishes tools for engineers and physical scientists. With better and more powerful tools we can do hard work easier. But that does not mean that each of us should own a private machine shop. That would be an economic waste. I have sometimes wished that I had a universal milling machine in my basement at home. But I would not buy one if I could. For the limited use I would make of it personally I could not take time to keep it in condition and myself in practice. But I know what such a machine tool can do, where to find one, and where to find men skilled in using it. Just so with mathematical tools. I can only afford to own and master a few—but I want to know what the others can do and where the men are who can use them.

Mathematics is simply an easier way of thinking about hard problems.

8. *Author's Closure on a Personal Note:*

A paper such as this is rarely published unless someone's grandson discovers a file of old notes in a cob-web covered trunk in the attic, and publishes them at his own expense to prove his granddad could read and write. These particular notes never would have seen the light of publicity in my lifetime had not the esteemed Secretary of our A.S.N.E. met me (by chance) and entered into one of those philosophic arguments which sailor-engineers are apt to start in such surroundings.

Therefore, some history of the notes on which this paper is based seems appropriate. When I was 10 years old I determined to be a Naval officer. When I was 13 years old I determined to be a scientific Naval officer and began to take notes

from books and papers which I read but did not own. I still have those notes, mostly transcribed on a typewriter later. I have continued that practice from the age of 13 on, nearly four-tenths of a century. The notes cover all sorts of subjects from anthropology to zoology, and range over more than 50 centuries of other men's thoughts. About 10 years ago, for a private purpose of my own, I sorted out *some* of the notes bearing on the subject of this particular paper and strung them together in much the manner they are presented here. It is that paper which I discussed with our Secretary and which, having been dressed up slightly, you see here.

I believe such studies have helped me in my career, even though my career has now ended in "physical disability incurred in line of duty." I know they have given me a great deal of happiness. I distinguish happiness from gaiety. You can have both together or either one alone without the other. I have had both.

If these notes help even one young engineer—I define a young engineer as one still to the windward of 40 years old—it will have been worth offering them for publication. If any other young engineers think I am just "preaching" then I ask them to perform the duty I have imposed on my children and my grandchildren under similar circumstances, namely to look me in the eye and say "No. 93."

No. 93 is one of Rochefoucauld's (63) maxims which the rising generation should frequently call to the attention of aging men. This time I will not translate, but will present it to you directly:—

"93. Les vieillards aiment à donner de bons préceptes pour se consoler de n'êtra plus en état de donner de mauvais exemples."

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PROPELLER BLOWERS.

By H. F. Schmidt.

Shortly after writing the paper on "Propeller Blowers" published in the February 1928 issue of the Journal of The American Society of Naval Engineers, the writer realized that the theory there presented for the action of propellers, applied not only to a thin bent section such as the blade sections used in the original experiments, but also to those of the type used as airfoils regardless of the proportions.

The result was that it appeared that the cavitation speed of screw propellers could be calculated by this method, as was shown in an article in the August 1933 issue of the American Society of Naval Engineers. It was there shown that the ordinary airfoil, or even such a distorted section as the type E propellers tested by Admiral Taylor could be calculated in the same way using the bi-secting angles of the inlet and outlet sections of the blade or wing to obtain the effective inlet angle, and the ratio of inlet to outlet pitch.

It was not until recently, however, that the writer has had available any data on blowers constructed with blades of different airfoil sections. These new tests have been checked by the original theory, and found to be in very good agreement with experimental results, using only data derived from the results published in the 1928 article with one exception, a correction for hub diameter. The original experiments had all been performed with propellers having a hub diameter equal to 33 per cent of the tip diameter and therefore the effect of hub diameter was not determined. The hub correction factor was obtained from tests by A. I. Ponomareff using the same blades and various hub diameters.

Since we are now not only interested in hubs of various diameters, but also in total pressure instead of static pressure, it may be advantageous to use larger inlet pitches and larger

ratio of pitch ratios, since the maximum efficiency for any given annular area increases with the capacity, that is, inlet pitch, up to much higher values than those for maximum static efficiency.

Consequently the formula for design in the 1928 paper which was for a hub diameter ratio 0.333 and ratio of pitch ratios of approximately 1.6 must be broadened, and corrections for projected area ratios other than .625 and ratio of pitch ratios other than 1.6 have to be provided for.

We will therefore use the general formula for volume per minute in cubic feet.

$$\text{Volume} = \text{annular area sq. ft.} \times \text{dia. ft.} \times \text{pr} \times \text{rpm} \times (\text{rpr corrections})$$

$$\text{Total head} = \frac{H_T}{C^2} \times (\text{projected area corr.}) \times (\text{hub corr.}) \times (\text{rpr correction})$$

The total head may be expressed in inches of water or foot pounds per pound of air or gas. If the head is desired in inches of water, H_T is the theoretical head in inches of water corresponding to the tip speed or if the result in foot pounds per pound of air is desired, $H_T = \frac{(\text{tip speed})^2}{2g}$

In Figure 1A the factor "C" is that in the equation

$$C = \sqrt{\frac{\text{theoretical head}}{\text{actual head}}}$$

with the exception that it has been corrected from static head used in the original article to total head which is more convenient in the design of multi-stage blowers.

Corrections for volume with variations of the outlet pitch divided by the inlet pitch or rpr, Figure 1B, is obtained from the tests of the propellers having ratio of pitch ratios less and greater than 1.6. A ratio of pitch ratios of 1.6 corresponds to the point of maximum efficiency.

Values of the correction to the calculated pressure for variations in projected area ratio was obtained from the tests with propellers having the same inlet pitch and ratio pitch ratios but with various projected area ratios and is given in Figure 2. The

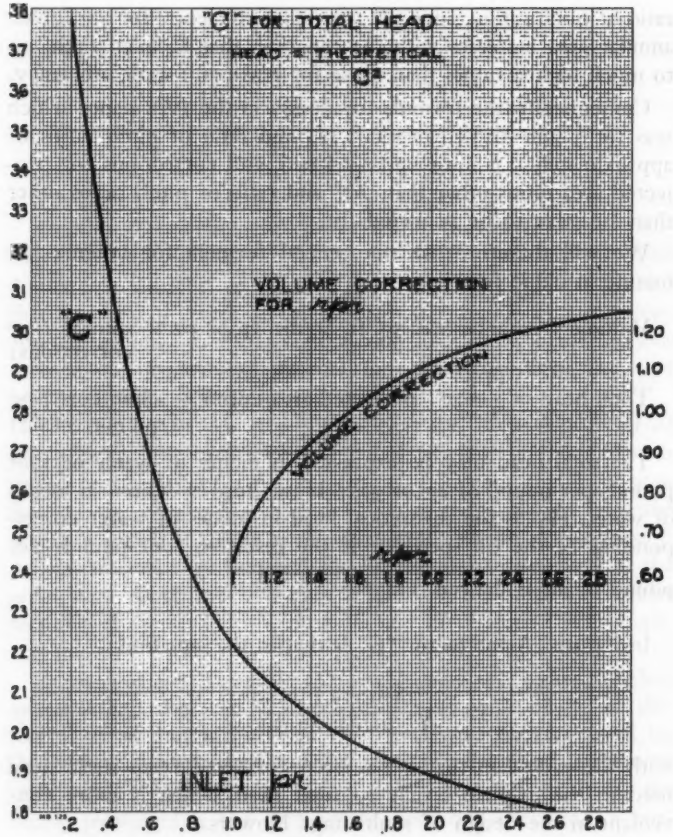


FIGURE 1-A AND 1-B.

pressure obtained for a projected area ratio of .625 is taken as unity, as it is the most efficient area ratio.

Variations of pressure for ratios of pitch ratio other than 1.6 which is taken as unity appear in Figure 3.

The only data taken from recent tests is Figure 4 showing the effect of hub diameter ratio, on pressure.

In the formulas for volume and head, the projected area and

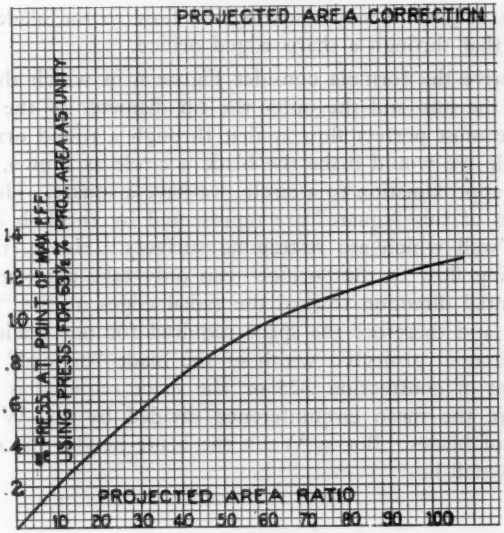


FIGURE 2.

rpr corrections were obtained from the early tests of propellers having different projected areas and constant inlet and outlet pitches. The rpr correction was obtained from tests of propellers of constant projected area and constant inlet pitch and various outlet pitches.

For use in the more general formula the correction factors for inlet pitch, projected area and ratio of pitch ratios, have been cross plotted from the tests given in the 1928 paper and are presented in Figures 1 to 3 and the hub correction is shown in Figure 4.

By selection of the required values from the four design curves and inserting these in the volume and pressure formulas given, the pressure and volume at the point of maximum efficiency corresponding to tangential inlet to the blades is obtained.

There still remains one design variable which is required, though not of such great importance with hubs of large diameter ratio—the optimum number of blades. This can be obtained

from the original theory assuming the flow to the blade as the same as that through a sharp edged orifice.

To derive the formula consider Figure 6 in which the inner circle is the hub and the outer the tip diameter of the blades. Now the most perfect flow is that through a round orifice, so if possible we would make the blade a circle as shown, and as the best approximation to a circle, we have a square. However, two sides of this "square" are represented by the hub and the housing wall which are restraining surfaces. Hence, the free contracting flow to the "orifice" can only take place in a circumferential direction toward the radial edges of the "square" or blade and therefore the space required for one blade is the circumferential projected width of the blade "divided" by the coefficient of contraction .625 or, the space required is 1.6 W. Therefore the mean circumference divided by 1.6 W is the number of blades. But as W=h the blade height, the number of blades will be—

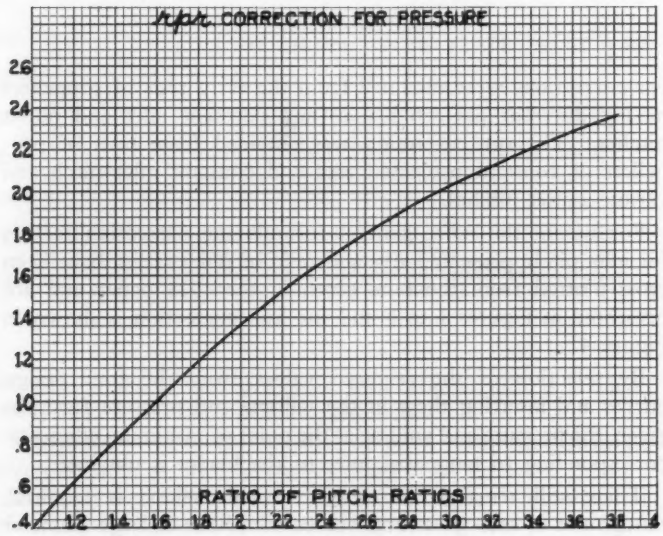


FIGURE 3.

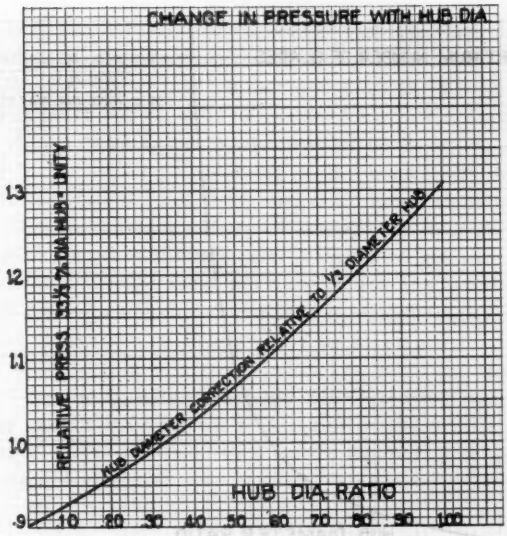


FIGURE 4.

$$n = \frac{(D_1 + D_2)}{1.6\ (D_1 - D_2)} = \frac{2\ (1 + \%)}{(1 - \%)}$$

where $\% = \frac{\text{hub dia.}}{\text{tip dia.}}$

This is plotted in Figure 5 and is in very good agreement with tests. As may be expected, as the hub gets larger and the number of blades increases, considerable variations from the optimum can be made without material changes in efficiency.

As a check of the application of the original data to airfoil sections, calculations were made for the point of maximum efficiency obtained on tests of 15-inch tip diameter blower, single stage, having various hub diameters and fitted with blades of N.A.C.A. section 6512, shown in Figure 7.

The blades have a chord length of 1¾ inches and are twisted 15 inches in their length.

The effective inlet and outlet angles of the blade were obtained

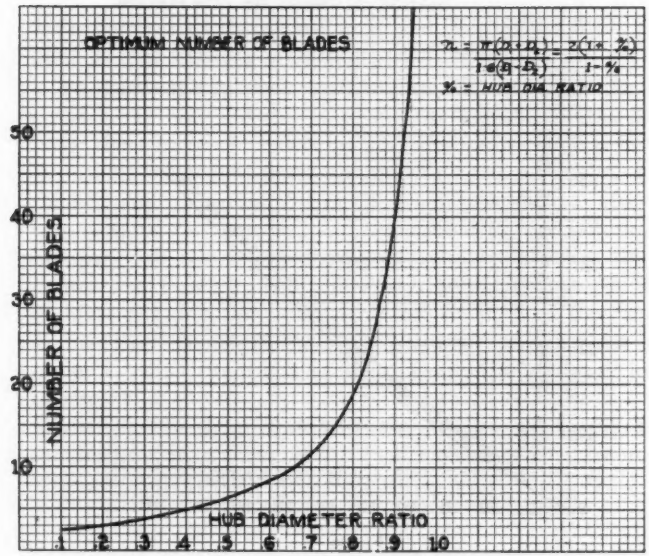


FIGURE 5.

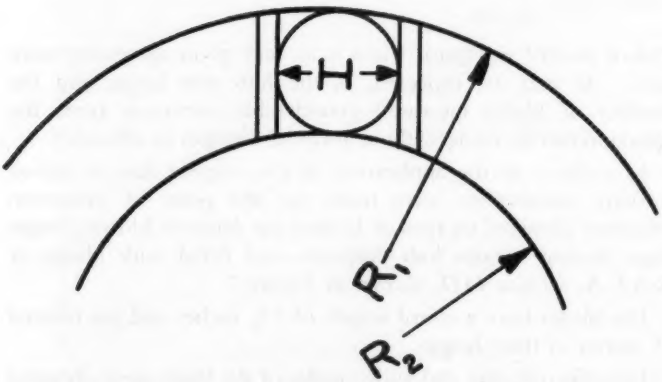


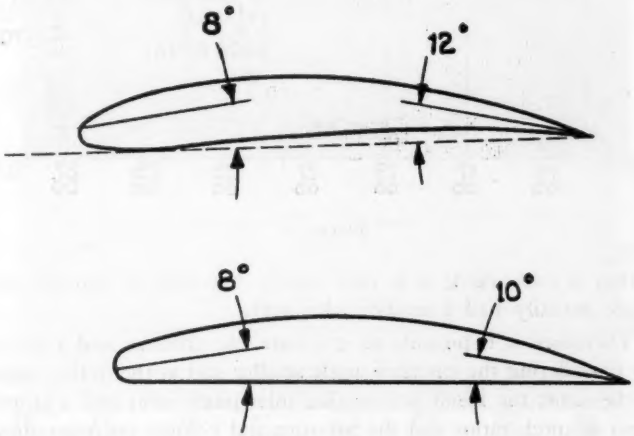
FIGURE 6.

by bi-secting the inlet and outlet sections of the blade as indicated, giving an inlet angle of 8 degrees smaller and an outlet angle 12 degrees larger than the face chord angle. Consequently these angles were subtracted from and added to the face chord angle to obtain the inlet pitch and ratio of pitch ratios.

Owing to the fact that the twist of the blades was not such as to give a constant inlet pitch and a constant ratio of pitch ratios at all radii, it was necessary to average the inlet and outlet sections at the inner and outer radii in order to get the average inlet pitch and average ratio pitch ratios for the whole blade. The pitch ratios so obtained were used in calculating the points on the test curves shown in Figures 8 to 11 inclusive.

When a blower is operating below its point of maximum efficiency, that is at reduced volume, the fluid no longer enters the blades tangentially but in a direction more nearly parallel to the plane of rotation. It thus enters the blades at an angle to the entrance edge.

Now, if the angle of impact with the blade is not too large, the fluid is turned very nearly as efficiently as though it entered tangentially. Consequently, when a blower is operating below its point of maximum efficiency or partial load, as far as the fluid



FIGURES 7 AND 12.

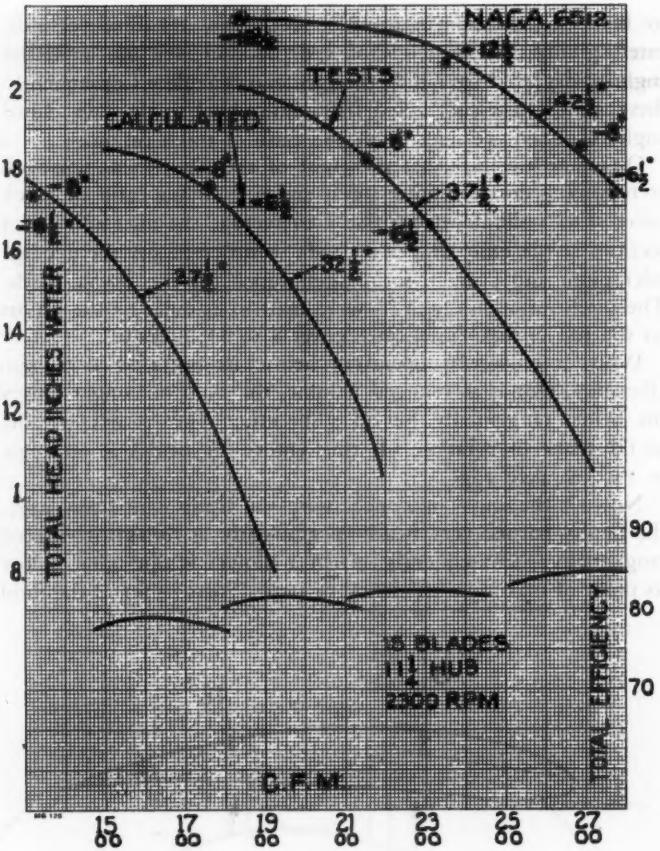


FIGURE 8.

action is concerned, it is very nearly the same as though the blade actually had a smaller inlet angle.

Therefore it is possible to calculate the pressure and volume by considering the entrance angle smaller and as the outlet angle is the same, the result is a smaller inlet pitch ratio and a larger ratio of pitch ratios and the pressure and volume corresponding

to these new values may be calculated in the regular way, giving additional points on the pressure volume curve.

In this manner the pressure volume curve may be extended to about three-quarter load, and by assuming that the inlet angle is *larger* than it actually is, the curve may be extended about ten per cent beyond the maximum efficiency point.

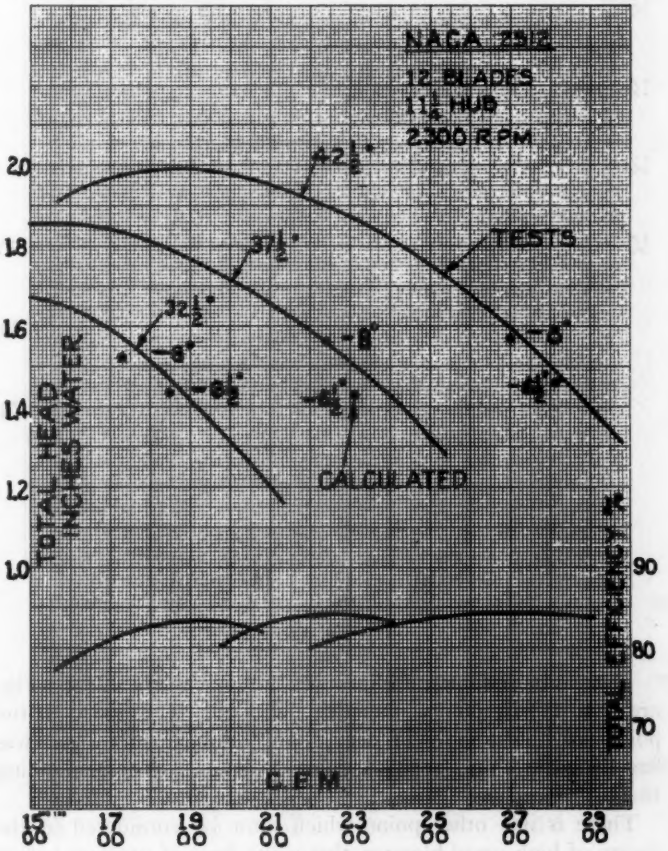


FIGURE 9.

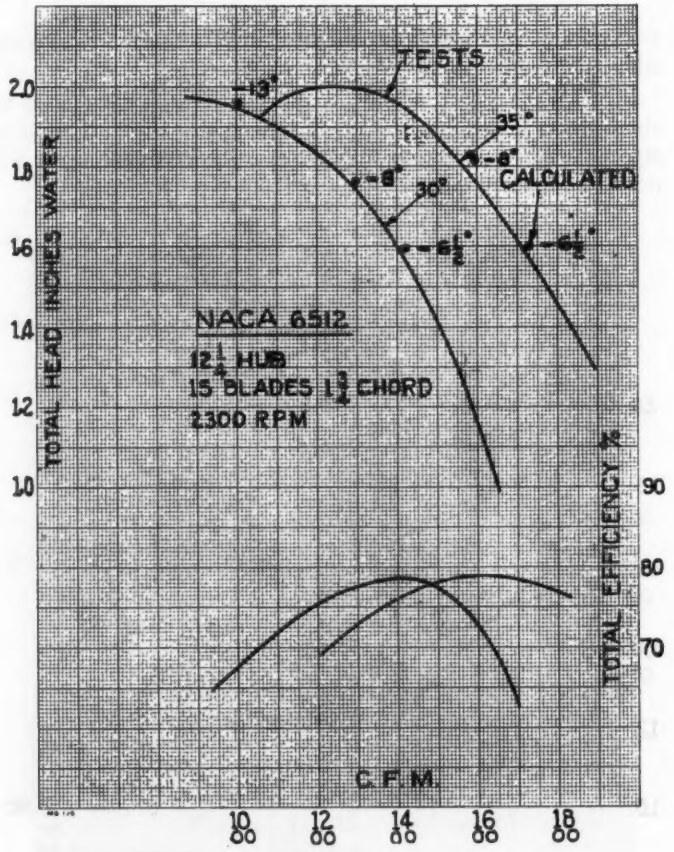


FIGURE 10.

This has been done and is indicated on some of the curves (Figures 8 to 11) by points other than full load. The figures at the points are the number of degrees the assumed inlet angle was less than the chord angle—8 degrees being that corresponding to normal capacity.

There is one other point which must be considered in the design of high speed blowers, that is the drop of pressure before

entrance to the first row of blades, since it is necessary generally to have a rather high axial velocity through the area swept by the first row of blades so that the pressure at the entrance of the first row of blades is not atmospheric but may be considerably below, and the specific volume of the entering air is also greater, reducing the weight of air handled.

Since the compression ratio created by given blading at con-

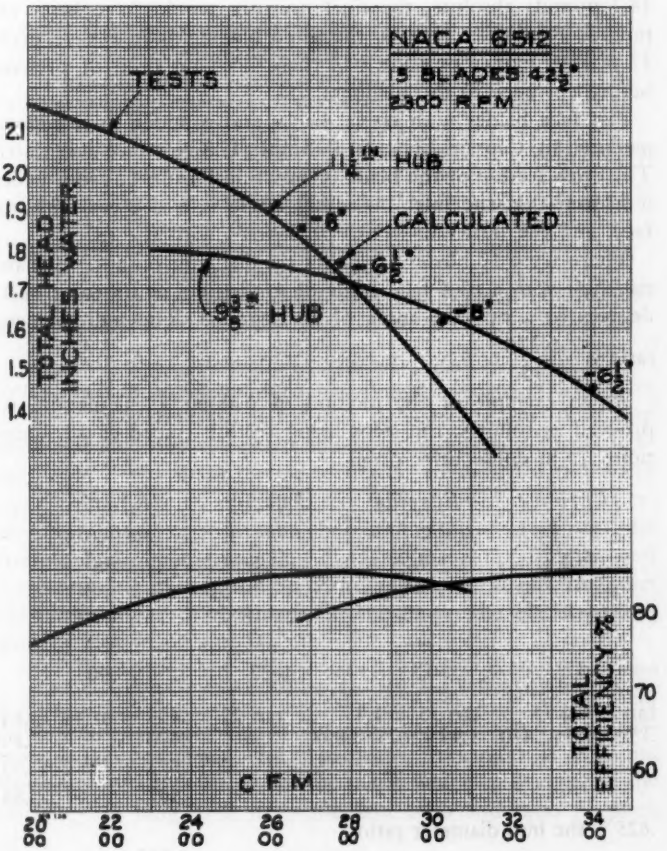


FIGURE 11.

stant speed is fixed for any assumed entrance velocity, the final pressure is directly proportional to the pressure at the inlet to the first row.

Thus, if the entrance velocity is sufficient to reduce the pressure from atmospheric to say 13 pounds absolute, and the compression ratio of the blower were 10., the final pressure would be 130 pounds absolute, whereas had the inlet pressure been 14.7 pounds absolute, the final pressure would have been 147 pounds absolute. That is, the same blading would have given 17 pounds higher pressure had there been no drop of pressure before entrance to the blading.

Flat face blades with ogival backs and rounded inlet edge as patented by Captain Lybrand P. Smith, U.S.N., were also tested. The cross section of the blade is shown in Figure 12 from which it is seen that the inlet bi-secting angle is 8 degrees with the face, and the outlet bi-sector 10 degrees with the face.

There were 12 blades 15 inches tip diameter on a 9.375 diameter hub with a root face angle of 45 degrees and a twist of 15 degrees or a tip face angle of 30 degrees. The hub diameter ratio is $\frac{9.375}{15}=.625$.

As an example of the method of making a calculation, we proceed as follows, the steps being readily traced as all operations are in consecutive order.

Effective inlet base angle	*Effective outlet base angle*	*Effective inlet tip angle*	*Effective outlet tip angle*
45	45	30	30
− 8	+10	− 8	+10
37°	55	22	40

tan 37°=.75 tan 55°=1.43 tan 22°=.403 tan 40°=.84

.75×.625=.470 1.43×.625=.895

.470 .895

2) .873 2) 1.735

.437 .867

.625 is the hub diameter ratio

Average pr=.437×π=1.375 Average rpr=$\frac{.867}{.437}$=1.98

To obtain the projected area ratio we take the mean projected width of the blade 1.27 inches $\times$ 12 blades = 15.2 inches and divide by the mean circumference of the blade circle 38.2, which gives 40 per cent. The pr, rpr, projected area ratio, rpm and the annular area, .747 square feet, swept by the blades is all the data needed to obtain the capacity and total head at the point of maximum efficiency from the formulae previously given.

V = area $\times$ dia. $\times$ pr $\times$ rpm $\times$ rpr correction (from Figure 1B)

$$h = \frac{H_T}{0^2} \times \text{P.A. (correction Figure 2)} \times \text{hub correction (Figure 4)} \times \text{rpr correction (Figure 3)}$$

$$H_T = \frac{(\text{tip speed ft. sec.})^2}{(66.7)^2} = 5.08 \text{ in inches of water}$$

Vol. cfm = .747 $\times$ 2300 (Rpm) $\times$ 1.25 (dia.) $\times$ 1.375 (pr.) $\times$ 1.11 (rpr) = 3280 C.F.M.

$$h = 5.08 \frac{1}{(2.045)^2} \times 1.33 \text{ (rpr)} \times .72 \text{ (PA)} \times 1.12 \text{ (hub)} = 1.32''$$

The test gave Volume = 3265 C.F.M.
Pressure = 1.35 inches

In conclusion, the facts which should be pointed out and observed, are these—

The method of calculation based on the "hole" theory has been shown to apply not only to propeller pumps and blowers but also to marine propellers for determining the cavitating point, but also that the performance of pumps or blowers having blades of any shape can be accurately predicted without any experimental data on the blade section employed. All the performance characteristics are defined for a given stage by inlet pressure and temperature, hub diameter ratio—tip diameter and revolutions per minute and tip chord angle and the effective inlet and outlet angles of the blades as shown on the blade sections shown in Figures 7 and 13.

Also the method and original data have been amply confirmed as indicating that the maximum efficiency occurs when the fluid enters tangentially, and when the ratio of pitch ratios is approximately 1.60 combined with a projected area ratio of .625 and no change of blade shape can be made which will improve the efficiency so long as it has same area ratio and ratio of pitch ratios.

The outstanding fact, however, is that this is the only method known to the writer by which a blower can be designed to give any possible required pressure and volume, the inlet pitch, outlet pitch, projected area ratio, tip and hub diameters being determined first, and then a blade section designed to meet the requirements, without reference to any known tests on a similar section, or any necessity for running any.

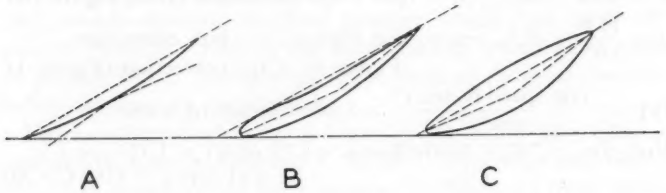


FIGURE 13.

Figure 13 illustrates how blade sections can be built up around any inlet and exit angles selected to meet the design requirements. The blade sections shown all have an inlet pitch ratio of 1.3 and ratio of pitch ratios of 1.70, giving the same pressure and capacity with decreasing efficiencies from A to C but increasing greatly in mechanical strength. The chord angles are not all the same.

It was during a lecture by Comdr. William Ledyard Cathcart, U.S.N. (retired), Professor of Marine Engineering at Columbia University in 1902, that he made the statement that the thrust of a propeller increased almost directly proportional to the projected area ratio up to a little over 60 per cent, but that beyond this it suddenly fell off and except to prevent cavitation, the little gain in thrust was not justified because of the falling off of efficiency.

The writer does not remember anything else Comdr. Cathcart said that day, because the thought flashed through his mind that the 60 per cent area and falling off in efficiency was related to the coefficient of contraction of a sharp edged orifice, the blade representing the orifice, and the dropping off of thrust and efficiency must be the result of interference between blades in preventing full flow to the blades when the area is too great.

After the lecture the writer mentioned this to Professor Cathcart and he received it in a very kindly and fatherly manner, but said he thought it just a coincidence. However, the thought would not go, but grew, and it is at last believed to have been justified.

ACCURACY OF SHAFT HORSEPOWER DATA.

By Harold L. Coffin (Civil Member).*

The question often arises as to the accuracy of shaft horsepower data. This is a natural question and quite in order, because comparison of performance of different types of machinery, and also the machinery performance in relation to contractual guarantees, are based upon shaft horsepower. Therefore, any errors in horsepower determination will result in errors in the analysis of data.

The following discussion is based on the methods and apparatus in use at the Navy Yard, Philadelphia, for calibrating torsion meters and computing the probable error.

Of the many types of torsion meters in existence, it is believed that the outside electric type is the most frequently installed. In one make of this type, the angular twist of the propeller shaft transmitting torque causes the air gaps of a differential transformer (attached to the shaft) to become unequal. This inequality (by transmission of current) displaces a galvanometer needle from its zero position. A numerically inscribed indicator drum, connected to a like transformer, is rotated until the galvanometer needle returns to zero. The resultant drum reading is a function of the torque transmitted. The shaft horsepower is computed from the formula:

$$SHP = \frac{RPM \times \text{meter reading}}{\text{meter constant}} \quad \ldots\ldots (1)$$

Each quantity in this equation will contribute a certain amount of error to the result, because all measurements are subject to three kinds of errors; namely, constant or systematic errors, mistakes and accidental errors. (1)

Systematic errors are mostly due to imperfections in construction or adjustment of instruments, the "personal equation" of the

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(1) Engineering Mathematics, U. S. Naval Institute, Publisher.

observer, etc. In this study, the personal equation is ignored, because it is of a negligible order.

Mistakes or blunders are large errors due to carelessness and do not follow any law. In this investigation, mistakes are not considered.

Accidental errors are those whose causes are unknown and indeterminate and follow the law of chance. The mathematical theory of errors deals only with accidental errors.

The total propeller revolutions required to traverse the measured mile is determined by starting and stopping an electric printing counter at the beginning and end of the mile. The total revolutions, divided by the time in minutes, gives the RPM. This, of course, is an average figure and does not indicate the amount of variation of the propeller revolutions. This variation, however, may be determined by means of a chronograph. It is understood that the probable error of the RPM is not greater than 1/10 of 1 per cent.

The torsion meter indicator drum is read ten times at nearly equal intervals over the measured mile. The arithmetical mean of these readings is used as the meter reading. Analysis of a number of meter readings demonstrated that the probable error was not greater than 2/10 of 1 per cent.

The third quantity in equation (1), i.e., the "meter constant" is found either by calibration or computation. The most accurate method is by calibration and the probable error by this method will be discussed first.

Calibration essentially consists of holding one end of the shaft in a fixed position and applying a twisting moment to the opposite end. As the twisting moment (or torque) is applied, angular deflection of the shaft occurs and is measured by the torsion meter. The meter readings, in conjunction with the twisting moment, allow computation of the meter constant.

In 1940, a portable calibration apparatus was designed and built by the Navy Yard, Philadelphia, that would accommodate shafts from about ten feet to fifty-eight feet in length and from about eight inches to twenty-three inches in diameter. A total load of forty thousand pounds at a radius of thirty feet, resulting in a twisting moment of 1,200,000 pounds feet, can be applied.

Figure 1 is a view of the apparatus erected to calibrate a stern tube shaft. The general arrangement of torsion meter, pointers and bearing stand is shown. The distance from the floor plates to the centerline of the shaft is nine feet, eight inches. The radius from the shaft centerline to either end of the torque arm is fifteen feet. The scales for measuring the load on each end of the torque arm are of forty thousand pounds capacity. The pointers allow measurement of angular deflection on a radius of ninety-two inches by means of a micrometer.

Figure 2 is a close-up showing the torsion meter, anti-friction bearings, method of securing the torque arm to the shaft and the upward load scale connection. The anti-friction bearings were especially designed for this apparatus. The rollers are ten inches in diameter with a twelve inch radius on the contact face. The complete bearing was designed for a maximum allowable load of 16,000 pounds. The bearing stand was fabricated from two inch IPS steel pipe with 1½ inch IPS pipe bracing, welded construction and stress relieved and tested with the adjustable plate in the maximum position with a static load of 24,000 pounds.

Figure 3 shows the method used to apply the downward load to the torque arm. As stated before, the scale is of 40,000 pounds capacity and is calibrated with known weights just prior to use.

Figure 4 is the specially designed micrometer that allows measurement of pointer movement in order to determine the modulus of rigidity. The micrometer has a travel of five inches with a least count of .001 inch or, in other words, it is possible to determine angular deflection of the shaft within 1/1800 of a degree.

In calibrating the torsion meter, the total twisting moment applied is generally equivalent to the torque produced by the main propelling machinery developing 10 per cent above rated full power. Expressed algebraically:

$$M_t = \frac{63030\,H}{n}\,1.1 \qquad (2)$$

where

M_t = twisting moment, inch-pounds
H = contract shaft horsepower
n = contract propeller RPM

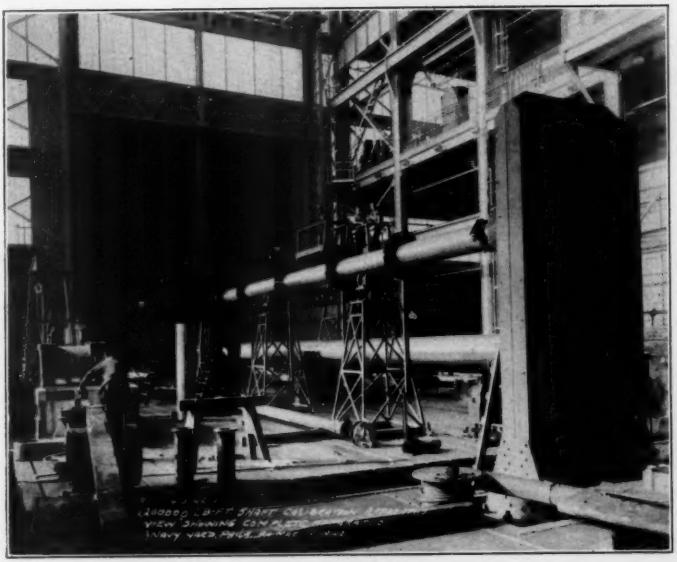

FIGURE 1.

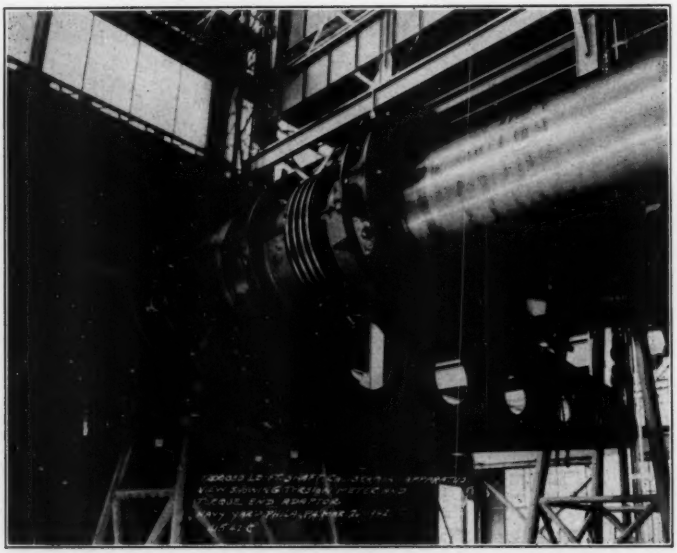

FIGURE 2.

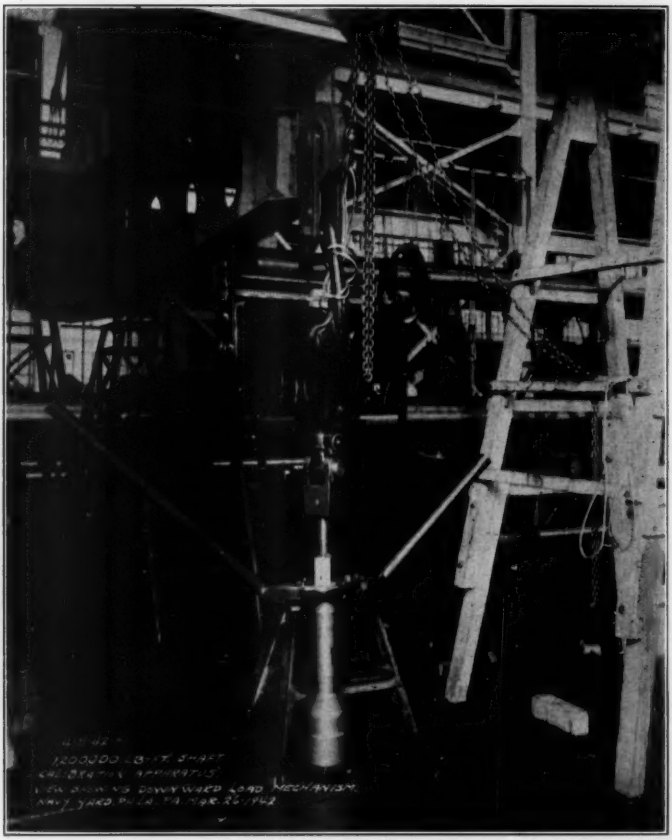

FIGURE 3.

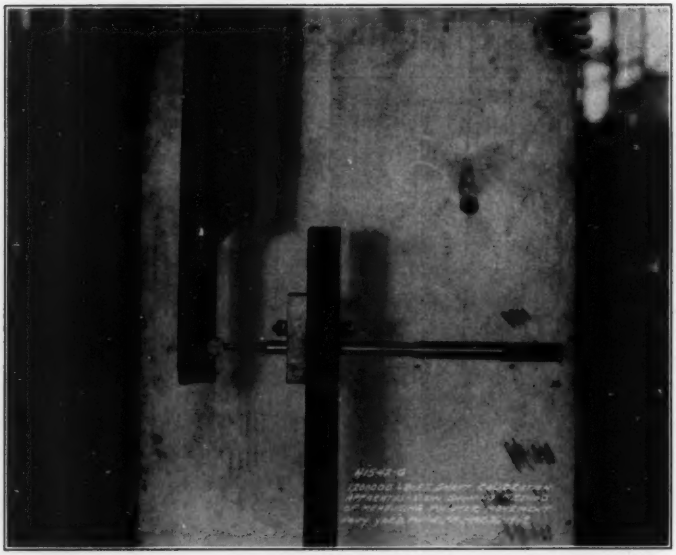

FIGURE 4.

The torsion meter constant by the calibration method is obtained by the formula:

$$\text{Constant} = \frac{33000}{2\pi Q} \qquad (3)$$

where Q = ft. lbs. torque per unit meter reading.

In calibrating the torsion meter, the meter reading is recorded for each increment of torque. The increment of torque is

CALIBRATION DATA			
TORQUE ARM LOAD POUNDS	TWISTING MOMENT FOOT-POUNDS	TORSION METER READING	TORQUE PER UNIT METER READING
10607	159100	54.8	2903
14297	214450	74.6	2875
18162	272430	95.6	2850
21752	326280	115.0	2837
17132	256980	91.5	2808
12967	194500	68.8	2827
9257	138850	48.5	2863
Q = Arithmetical mean = 2852			

$$r_0 = 0.6745 \sqrt{\frac{v_1^2 + v_2^2 \ldots\ldots v_7^2}{n(n-1)}}$$

$$r_0 = 0.6745 \sqrt{\frac{6041}{42}}$$

$$r_0 = 8$$

$$\%\, r_0 = \frac{8}{2852} \times 100 = 0.3$$

probable error of $Q = \pm 0.3\%$ plus $\pm 0.5\%$ torque arm & weight error

$$Q = 2852 \pm 0.8\%$$

$$\text{Torsion Meter Constant} = \frac{33000}{2\pi Q}$$

$$= \frac{33000}{6.283 \times 2852}$$

$$= 1.842 \pm 0.8\%$$

FIGURE 5.

divided by the meter reading. This reduces all readings to a common basis and allows the probable error to be computed. However, in order to determine the total error of the meter constant, the torque error must be determined by measuring the torque arm and weights. This error (known as the load error) at the Navy Yard, Philadelphia, is of the order of 0.5 per cent.

At the Yard, the probable error of the calibrated torsion meter constant, with all corrections applied, has not been greater than 1.6 per cent nor less than 0.6 per cent.

A typical set of calibration data, together with computation of the probable error of the meter constant, is shown in Figure 5.

Returning to formula (1), the probable error of SHP determination with an error of 0.1 per cent in RPM, 0.2 per cent in meter reading and 1.6 or 0.6 per cent in the meter constant would be as follows:

$$\text{SHP} = \frac{\text{RPM} \pm 0.1\% \times \text{meter reading} \pm 0.2\%}{\text{meter constant} \pm 1.6\% \text{ or } 0.6\%} \qquad (4)$$

therefore the SHP would be correct to within 0.9 or 1.9 per cent.

The probable error without calibration of the torsion meters is much greater. This is demonstrated by analysis of the following formula:

$$\text{meter constant} = \frac{1605 \times 10^5 \times L \times B}{C\ (R^4 - r^4)} \qquad (5)$$

where L = Length between clamping planes of meter on shaft.
B = Radius of circle described by center of transformer poles on shaft.
C = Modulus of rigidity.
R = Radius of outside diameter of shaft.
r = Radius of inside diameter of shaft.

The constant 1605×10^5 will be assumed to have no error. The error of L is approximately 0.1 per cent and B approximately 0.3 per cent if careful measurement to the nearest 1/32 inch is followed. The error of R and r is negligible if carefully measured to 0.001 inches.

The modulus of rigidity C has the greatest error. The modulus for AN steel is generally assumed to be 11,600,000. The calibrated modulus of rigidity for more than fifty shafts showed a variation

of approximately four per cent from 11,600,000. If the modulus is assumed, it is possible, then, to have a four per cent error.

Reducing the meter constant formula to per cent error, we find

$$\text{meter constant} = \frac{1605 \times 10^6 \times L \pm 0.1\% \times B \pm 0.3\%}{C \pm 4.0\% \qquad (R' - r')} \quad \ldots\ldots (6)$$

or the meter constant by computation would have an error of approximately 4.4 per cent.

The foregoing has been written with the intent of creating an interest in investigating and determining the probable error of shaft horsepower data. It is believed that a continuing study of the accuracy of SHP data would be valuable to the profession.

FUEL OIL CAPACITIES OF SPRAYER PLATES.

By Fred J. Wiegand*.

The present discussion deals with a chart and table for the ready determination of sprayer plate capacities. Since these values are dependent upon several variable conditions, a review of the pertinent factors involved is considered desirable.

Mechanical pressure-atomizing fuel oil burners require an atomizing tip or sprayer plate. The function of this device is to break up the fuel into small particles and to distribute those particles so that they can readily mix with air for efficient combustion. In the case of the one-piece sprayer plate as manufactured by the Philadelphia Navy Yard and by several burner manufacturers, this is done by supplying the fuel under pressure through tangential slots or grooves into one end of a semi-spherical chamber (whirling chamber) at the other end of which there is an exit orifice. The quantity of fuel oil which can be atomized by such a sprayer plate is dependent upon the viscosity of the fuel oil, fuel oil pressure, area and length of the tangential slots, size of whirling chamber, area of orifice, and length of orifice. The sprayer plate as used in the Navy usually is provided with four or six tangential slots, each of the same cross-sectional area. If the ratio of total slot area to orifice area is 0.8, the sprayer plate is known as an 08 ratio plate. Each sprayer plate is designated by four digits, the first two indicating the twist drill size used in making the orifice and the last two the ratio.

Nearly all sprayer plates in Naval use are ⅞ inch outside diameter and the present discussion deals only with this type. Plates of this dimension having a whirling chamber 0.281 inch in diameter are known as standard when used with a nozzle having a boss 0.500 inch in diameter. The effective slot length, as determined by the diameter of the solid nozzle boss against

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* Associate Mechanical Engineer, Naval Boiler and Turbine Laboratory.

Size	Constant Standard Dish	Constant Standard Flat	Constant S. W. R. Dish	Constant S. W. R. Flat
1808			1.91	
1908			2.13	
2008			2.40	
2108			2.52	
2208			2.66	
2308			2.86	
2408			3.03	
2508			3.24	
2608			3.51	
2708			3.81	
2808	6.89	8.05	4.31	3.51
2908	7.52	8.84	4.69	4.16
3008	9.66	11.13	5.90	5.42
3108	12.31	14.20	7.78	7.30
3208	14.13	15.85	9.39	8.62
3308	15.48	17.32	10.28	9.71
3408	16.54	18.16	11.48	10.46
3508	17.32	19.13	11.81	10.87
3608	19.91	21.85	13.26	12.79
3708	21.40	23.84	14.85	14.14
3808	23.93	26.23	16.20	16.02
3908	26.02	28.29	17.95	17.74
4008	26.95	29.36	19.13	18.68
4108	29.60	31.69	20.53	20.70
4208	33.26	35.93	22.88	23.29
4308	39.14	41.95	27.89	28.74
4408	44.61	47.04	32.59	34.43
4508	54.38	56.33	39.81	42.87
4608	57.00	59.10	42.07	
4708	66.16	68.35	48.50	

Size	Constant Standard Dish	Constant Standard Flat	Constant S. W. R. Dish	Constant S. W. R. Flat
1809			1.45	
1909			1.59	
2009			1.91	
2109			2.07	
2209			2.17	
2309			2.34	
2409			2.52	
2509			2.84	
2609			2.98	
2709			3.14	
2809	5.89	6.74	3.60	
2909	6.49	7.42	4.01	
3009	8.21	9.34	5.21	
3109	10.60	11.91	6.69	
3209	12.09	13.55	7.78	
3309	13.41	15.04	8.62	
3409	14.21	16.14	9.39	
3509	14.91	16.54	9.82	
3609	17.16	19.07	11.45	
3709	18.76	20.67	12.42	
3809	20.88	23.07	14.00	
3909	22.82	24.75	15.39	
4009	23.67	25.62	16.02	
4109	25.62	27.84	17.27	
4209	29.12	31.42	19.69	
4309	34.63	36.78	23.83	
4409	40.91	41.95	26.55	
4509	48.06	50.22	32.95	
4609	49.94	52.54	35.02	
4709	58.39	60.94	40.79	

2810	5.11	5.74	3.01	
2910	5.69	6.21	3.37	
3010	7.27	7.97	4.49	
3110	9.24	10.19	5.85	
3210	10.55	11.70	6.74	
3310	11.76	12.94	7.59	
3410	12.81	13.97	8.33	
3510	13.15	14.29	8.57	
3610	15.26	16.54	10.17	
3710	16.64	18.10	10.94	
3810	18.45	19.91	12.42	
3910	20.18	21.93	13.67	
4010	20.81	22.64	14.39	
4110	22.65	24.47	15.73	
4210	25.92	28.06	17.95	
4310	30.36	32.82	21.15	
4410	34.79	37.30	24.61	
4510	42.37	44.15	30.42	
4610	44.38	46.04	32.24	
4710	51.94	53.76	36.65	
4810			40.79	
4910			48.18	
5010			55.36	
5110			60.51	
5210			69.16	
5310			83.51	

2811	4.48	4.83		2.49
2911	4.91	5.32		2.84
3011	6.43	6.93		3.78
3111	8.21	8.84		5.01
3211	9.34	10.00		5.84
3311	10.53	11.27		6.59
3411	11.33	12.00		7.23
3511	11.70	12.38		7.50
3611	13.52	14.45		8.83
3711	14.82	15.81		9.82
3811	16.54	17.48		11.24
3911	18.04	19.32		12.30
4011	18.69	19.84		12.96
4111	20.39	21.47		14.19
4211	23.23	24.66		16.81
4311	27.17	28.88		20.44
4411	31.02	32.82		24.16
4511	38.39	40.30		29.64
4611	39.91	42.37		31.73
4711	47.04	49.12		37.51
4811				40.05
4911				42.60
5011				48.50
5111				55.77
5211				68.04
5311				82.03

which the sprayer plate is held in the burner barrel, is thus 0.1095 inch. In order to obtain a greater range of capacity by varying the fuel oil pressure, a nozzle boss of 0.313 inch diameter is sometimes used, by this means reducing the effective slot length to 0.016 inch. When a standard sprayer plate is used with a nozzle boss of this reduced diameter, the arrangement is designated as "semi-wide range." Because of the reduction in slot length, the capacity of any sprayer plate is greater when used with the semi-wide range (S.W.R.) nozzle than when used with the standard nozzle, all other conditions remaining the same. The face of the sprayer plate in which the orifice terminates is either finished off flat or dished. If flat, the orifice will have a designed length of 0.070 inch. If dished, the orifice will be rounded off with an effective length of about 0.020 inch. It is thus apparent that a flat-faced sprayer plate will have a capacity different from that of a dish-faced one.

From the results of many boiler tests, the Naval Boiler and Turbine Laboratory has determined the actual capacity of the sizes of sprayer plates most commonly used in Naval boilers. Oil having a viscosity of 150 second Saybolt Universal (SSU) was used in all cases. The values were plotted on logarithmic co-ordinate paper for pressures up to 300 psi, the graphs for any particular ratio being shown on a single sheet. Of course, four separate sets of curves had to be drawn on as many sheets for each different ratio because the same size sprayer plate could be either flat-faced or dish-faced and could be used with either a standard or a semi-wide range (S.W.R.) nozzle. There being eight ratios under consideration, a resulting collection of thirty-two sheets were required to record graphically the relationship between pressure and capacity of the sizes commonly used.

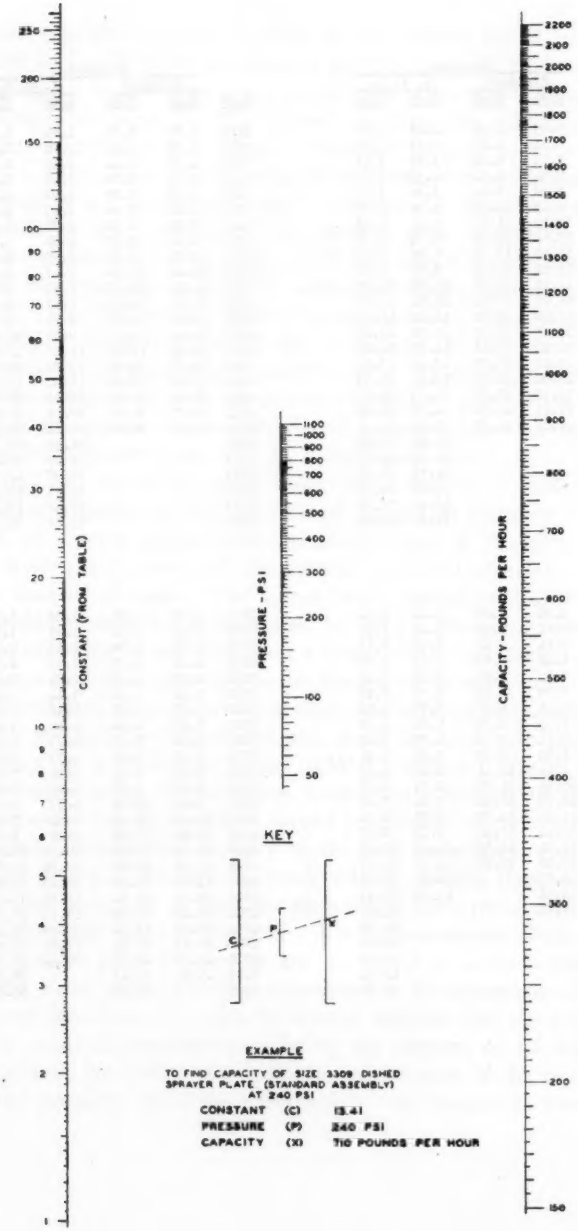
The object of the present work was to compile these data in a simpler form and at the same time make them more useful by extending the range of pressure. Taking cognizance of the fact that sprayer plate dimensions are permitted to deviate slightly, within close manufacturing tolerances, a consideration of the plotted data from 722 capacity curves showed that the average slope could be considered as being the tangent of 61 degrees 30 minutes, or 1.842. Accordingly, the relation of the capacity to the pressure could be expressed by the empirical formula,

Size	Constant Standard Dish	Constant Standard Flat	Constant S. W. R. Dish	Constant S. W. R. Flat
2812	3.93	4.34	2.28	2.25
2912	4.34	4.83	2.57	2.57
3012	5.74	6.26	3.37	3.45
3112	7.34	8.12	4.50	4.64
3212	8.38	9.24	5.26	5.39
3312	9.49	10.36	5.96	6.03
3412	10.23	11.13	6.44	6.48
3512	10.55	11.61	6.74	6.79
3612	12.22	13.33	7.96	8.10
3712	13.41	14.41	8.73	8.87
3812	14.91	16.19	10.02	10.05
3912	16.39	17.70	10.94	10.97
4012	17.00	18.28	11.45	11.48
4112	18.45	20.04	12.58	12.83
4212	21.10	23.41	14.59	14.85
4312	25.13	27.39	17.40	17.16
4412	28.88	31.83	19.21	21.33
4512	35.43	38.57	23.19	26.55
4612	37.13	40.30	24.73	28.74
4712	43.03	46.78	30.10	34.43
4812			33.87	37.51
4912			39.33	46.30
5012			44.53	55.77
5112			50.87	64.36
5212			60.97	75.17
5312			75.82	86.62

Size	Constant Stapdard Dish	Constant Stapdard Flat	Constant S. W. R. Dish	Constant S. W. R. Flat
2815	2.99	2.99	1.60	1.75
2915	3.33	3.41	1.78	2.00
3015	4.38	4.44	2.16	2.62
3115	5.69	5.79	3.45	3.45
3215	6.49	6.79	3.91	4.19
3315	7.34	7.56	4.47	4.68
3415	7.96	8.22	4.77	5.08
3515	8.21	8.52	4.96	5.26
3615	9.55	9.91	5.90	6.30
3715	10.40	10.92	6.47	6.80
3815	11.64	12.25	7.55	7.92
3915	12.81	13.55	8.27	8.69
4015	13.23	14.05	8.78	9.04
4115	14.49	15.35	9.61	10.02
4215	16.44	17.87	11.45	11.63
4315	19.71	21.17	13.81	14.24
(4415	22.65	24.66	16.02	16.55
4515	28.06	30.62	18.15	21.15
4615	29.73	32.25	20.44	22.08
4715	34.32	38.02	24.05	26.30
4815			26.55	29.79
4915			31.39	36.23
5015			36.87	42.33
5115			44.25	49.83
5215			57.42	58.71
5315			75.82	70.29

2818	2.34	2.25	1.20	1.44
2918	2.61	2.55	1.41	1.53
3018	3.41	3.41	1.90	2.12
3118	4.48	4.79	2.64	2.92
3218	5.23	5.54	3.12	3.40
3318	5.79	6.26	3.57	3.88
3418	6.26	6.80	3.91	4.16
3518	6.43	7.04	4.01	4.27
3618	7.64	8.38	4.78	5.11
3718	8.38	9.28	5.21	5.56
3818	9.44	10.61	6.07	6.43
3918	10.36	11.79	6.74	6.99
4018	10.73	12.31	7.06	7.46
4118	11.70	13.59	7.88	8.16
4218	13.44	15.35	9.22	9.47
4318	16.04	19.38	11.45	11.59
4418	18.88	22.90	13.35	13.57
4518	23.23		16.32	17.01
4618	24.38		17.20	17.95
4718	28.76			21.61
4818				24.16
4918				29.34
5018				35.62
5118				41.04
5218				50.87

2820	1.97	1.96	1.05	1.28
2920	2.35	2.22	1.25	1.50
3020	3.01	2.96	1.63	1.91
3120	4.04	3.93	2.33	2.68
3220	4.49	4.65	2.74	3.19
3320	5.11	5.32	3.14	3.54
3420	5.59	5.79	3.48	3.72
3520	5.89	6.04	3.64	3.94
3620	6.74	7.13	4.23	4.51
3720	7.49	7.80	4.73	5.11
3820	8.33	8.84	5.34	5.65
3920	9.03	9.77	5.82	6.23
4020	9.66	11.00	6.37	6.74
4120	10.40	11.21	7.01	7.41
4220	11.88	13.08	8.18	8.25
4320	14.49	15.90	10.46	10.31
4420	17.21	18.82	12.58	12.11
4520	21.10	23.49	15.43	14.80
4620	22.41	24.94	16.94	15.73
4720	25.42	29.98	19.45	18.90
4820	29.24	33.56	22.88	21.42
4920	35.43	39.52	27.89	25.80
5020	41.52	46.04	36.44	31.73
5120	49.94	53.45		38.18
5220		65.31		50.18


CONSTANT (FROM TABLE)
250
200
150
100
90
80
70
60
50
40
30
20
10
9
8
7
6
5
4
3
2
1
PRESSURE - PSI
1100
1000
900
800
700
600
500
400
300
200
100
50
CAPACITY - POUNDS PER HOUR
2200
2100
2000
1900
1800
1700
1600
1500
1400
1300
1200
1100
1000
900
800
700
600
500
400
300
200
150
KEY
C
P
X
EXAMPLE
TO FIND CAPACITY OF SIZE 3309 DISMED SPRAYER PLATE (STANDARD ASSEMBLY) AT 240 PSI
CONSTANT (C) 13.41
PRESSURE (P) 240 PSI
CAPACITY (X) 710 POUNDS PER HOUR

$P=kW^{1.842}$. The value of k for each sprayer plate size was found by substituting in this formula corresponding values of pressure (P) and capacity (W) as taken from the actual capacity curves. The values selected were at the center of the graph to reduce the effect of deviation from the average slope. The values of k for 652 different sprayer plates were thus calculated and are shown herewith in the table. The alignment chart here shown was then constructed on the basis of the empirical formula from which the capacity of standard sprayer plates with 0.281 inch diameter whirling chamber, using fuel oil having a viscosity of 150 SSU, can be obtained at any given pressure from 50 to 1100 psi—by applying the constant shown in the table.

METHOD OF ASSEMBLING MAIN REDUCTION GEARS.

By Frank E. Freeman*.

Several months ago an entirely new method of main reduction gear installation on U. S. Navy Destroyers was developed by the writer at the Bath Iron Works Corporation, Bath, Maine. This new method resulted in large labor saving and it is described here in order that other yards may benefit by its use.

Previously, this yard installed these reduction gears by the "piece by piece" assembly aboard ship. This new method differs from the old in that the entire upper case, including gears, pinions, and bearing caps, are all assembled at the place of manufacture and installed aboard ship without disassembly. The lower case installation is the same in the new as in the old method, that is, the case is chocked up in plane, oil pan, bull gear and main thrust assembled complete.

The present method is to assemble the entire first reduction and the second reduction pinions in the upper case, complete with bearings and bearing caps, before placing the upper case aboard ship. To date, 24 units have been installed by this method on Bath built Navy Destroyers with complete success. Two of these units were completely assembled at the place of manufacture and it is intended to have the units completed this way in the future.

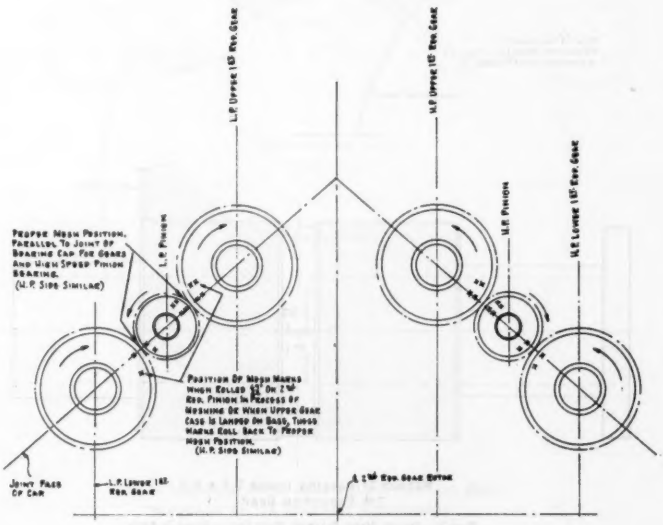
The lower case, the bull gear and its journal bearings, and the main thrust bearing with its caps are all assembled aboard ship in the usual manner. At the same time the upper assembly is made complete, except for covers, in the shop. Bridge gauge readings and end floats are also checked there.

To insure engaging in the proper position, two marks are made on each lower second reduction pinion while the assembly is still in the shop. With the proper aligning marks on the

* Mr. Frank E. Freeman, Quarterman, Machinery Division, Bath Iron Works Corporation.

forward ends of the first reduction pinions and gears all in line with the cover-joint faces, a mark is made on the barrel, between the helices of each lower-second reduction pinion, in line with the cover-joint face on that side. Then each lower second reduction gear is rolled up (the mark just described moves upward) so that a second mark may be inscribed in line with the cover-joint face 49/64 inch below the upper mark, measured on the barrel periphery. When contact between the lower second reduction pinion and the bull gear is about to be made as the assembly is lowered into place, the lower of these two marks should be in the plane of the cover-joint face.

The upper assembly is then lowered into the engine room by the dock crane. It is rigged with straps through the lightening holes in the upper case itself, fore and aft. In the forward engine room, the assembly must be tipped to clear the deck opening and to facilitate this, special lifting gear on each cover of the case allow the use of chain falls to do so. In the after

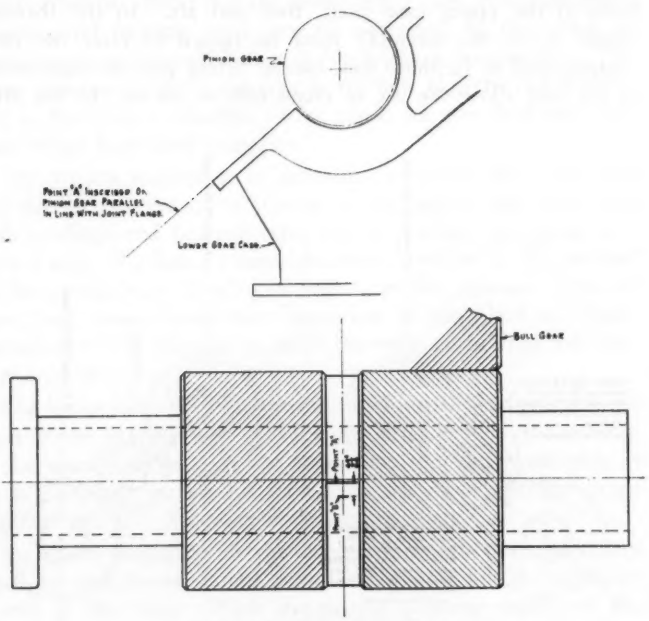

L.P. Upper 1st Red. Gear
H.P. Upper 1st Red. Gear
L.P. Pinion
H.P. Pinion
H.P. Lower 1st Red. Gear
Proper Mesh Position, Parallel To Joint Of Bearing Cap For Gears And High Speed Pinion Bearing. (H. P. Side Similar)
Position Of Mesh Marks When Rolled 49/64" On 2nd Red. Pinion In Process Of Meshing Or When Upper Gear Case Is Lowed On Bass, These Marks Roll Back To Proper Mesh Position. (H. P. Side Similar)
Joint Face Of Cap
L.P. Lower 1st Red. Gear
L.P. 2nd Red. Gear Bottom

SECTION - LOOKING AFT

FIGURE No. 1

engine room, the assembly is transferred to chain falls to carry it aft over its final position, using the special lifting gear on the overhead to which the chain falls are shackled. The assembly is lowered (in the forward engine room by the chain falls depending from the dock crane, in the after engine room by the chain falls depending from the overhead) with the aid of guide pins, to within one inch of its final position. Steel blocks one inch square by a foot long are placed on the lower case and the upper assembly on being lowered, rests on these blocks. This is to insure parallelism of the two joint faces. At this point all rigging is removed.

Four jacking screws are then used to take the weight of the assembly off the spacer blocks and the blocks are then removed,

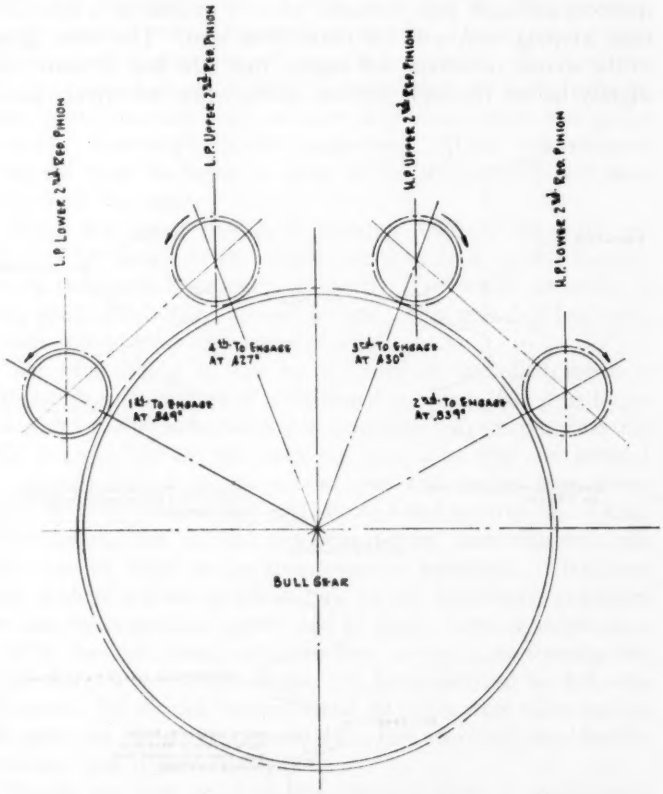


Method Of Marking Lower L. P. & H. P. 2nd Reduction Gear

Point 'A'- Proper Mesh Position When Case Is Joined To Base
Point 'B'- Starting Line. Follow Up Backlash Until Case Is On Base.

FIGURE No. 2

leaving the jacking screws supporting the upper assembly. For this purpose, the four regular 3/4 inch jack screw holes in the upper case corners are re-threaded to 7/8 inch standard and special jacking screws made up. These screws are made with a flat tip, case hardened, and have the head marked off in num-


L.P. Lower 2nd Red. Pinion
L.P. Upper 2nd Red. Pinion
H.P. Upper 2nd Red. Pinion
H.P. Lower 2nd Red. Pinion
4th To Engage At 427°
3rd To Engage At 430°
1st To Engage At 849°
2nd To Engage At 839°
Bull Gear

When Lowering Upper Gear Case In Position, Gears To Engage At The Indicated Dimensions.
Looking Forward.

FIGURE No. 3

bered quarters. They should be about 2 1/2 inches in length under the head.

With the weight of the upper assembly supported on the four jacking screws and the one-inch blocks removed, the assembly is then lowered by four men turning the jack screws in unison. The heads of the screws are marked off in numbered quarters and each man turns his screw a quarter of a turn at a time, keeping even with the other three men. The lower gears in the second reduction will engage first (the low pressure side slightly before the high pressure side), when the upper case is

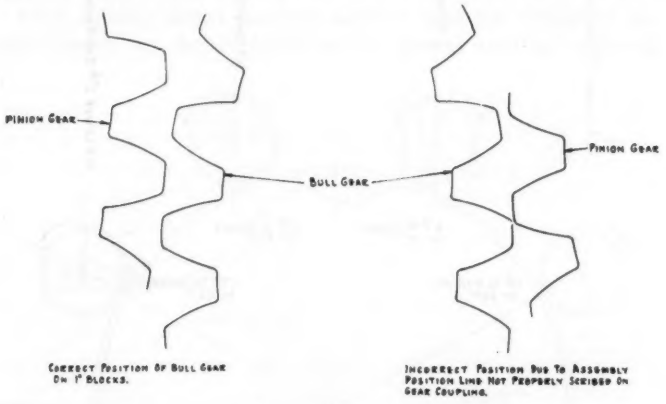

PINION GEAR
BULL GEAR
PINION GEAR
CORRECT POSITION OF BULL GEAR ON 1" BLOCKS.
INCORRECT POSITION DUE TO ASSEMBLY POSITION LINE NOT PROPERLY SCRIBED ON GEAR COUPLING.

FIGURE No. 4

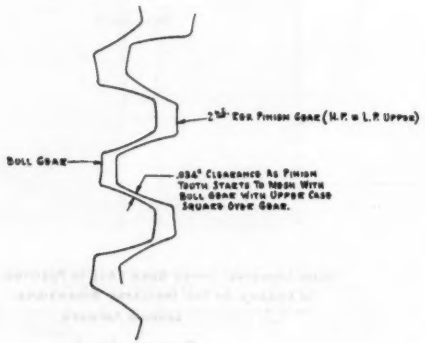

2ND ECH PINION GEAR (H.P. & L.P. UPPER)
BULL GEAR
.004" CLEARANCE AS PINION TOOTH STARTS TO MESH WITH BULL GEAR WITH UPPER CASE SQUARED OVER GEAR.

FIGURE No. 5

within 27/32 inch of the lower case, the uppers engage at approximately 7/16 inch.

Just before the uppers engage, they must be rolled to take advantage of the backlash in the train. From the time the lower gear engages the bull gear until it is in its final position, it rolls from a position such that the lower inscribed line on its barrel is in the plane of the cover-joint face to a position such that the upper line is in that plane. That is, such that the mating marks on the forward ends of the gears are in line. Due to the location of the upper, it must not roll, once it is engaged. To allow the upper to remain in its engaging position while the lower rolls, in spite of the fact that they are part of a locked train, the upper is rolled downward before engagement. Then the rotation delivered from the lower is taken up by the backlash and does not affect the upper.

When the upper case is in position, both lower gears are checked by means of the upper inscribed lines on the barrels, to see that each is properly engaged. The entire assembly is then ready for bolting up and covers. This assembly less main covers and end covers weighs 15 tons.

The total saving in time by the method described above is difficult to calculate but it is estimated at about 1000 man hours per unit or 2000 man hours per destroyer. From the time the gear is picked up on the dock the gear is in ship and meshed in proper position in about one hour and within four hours from the start the gear has been checked and covered up. Other departments, such as pipe coverers, pipers, tinsmiths, etc., are then free to work in the area formerly restricted. Thus, not only is there a great saving in time by the installation engineers but also by cranemen, riggers, and all trades working in the area.

With the unit closed within an hour or two after entering the ship, there is much less chance for loose material to fall into the gears. By the old method, about 40 hours were taken before the gear was closed up completely. This involved considerable overtime that is now eliminated.

Besides the time saved in the shipyard, there is considerable saving by the gear manufacturer since the unit may be shipped almost immediately after running final tests.

PRESSURE LOSSES IN MARINE FUEL OIL SYSTEMS.

By Cyrus Beck* and H. M. Miller†.

Note: The opinions or assertions contained in this article are the private ones of the writers and are not to be construed as official or reflecting the views of the Navy Department or the Naval service at large.

Introduction.

In order to obtain reliable ship design data where space and weight limitations impose restrictions on the pipe size to be employed for fuel oil service, the Bureau of Ships, Navy Department, in 1939, directed the investigation of pressure losses in fuel oil systems at the Naval Boiler and Turbine Laboratory. This work has been conducted under the direction of the Director, Captain C. A. Bonvillian, U.S.N., and the direct supervision of Mr. A. Howarth, Senior Chemist.

Fuel oil systems in marine installations differ from those in commercial plants in many respects and particularly in the number of fittings employed in comparison with the amounts of straight pipe. The customary method of including the pressure losses chargeable to valves, fittings, bends, etc., is to add to the straight pipe the equivalent length of each unit which sum then is used in formulas for calculation of the total pressure drop of the system. Accordingly, the value of equivalent length assigned to a unit assumes considerable importance in the design of fuel oil piping for marine purposes, where the total equivalent length of the fittings, etc., may be much greater than the actual lengths of the pipe. Unfortunately, the reported equivalent lengths of fittings, etc., in the laminar region have varied widely.‡ Further,

* Assistant Mechanical Engineer, Naval Boiler and Turbine Laboratory.
† Formerly connected with the Naval Boiler and Turbine Laboratory.
‡ The Design of Fuel Oil Piping for Marine Purposes by John A. Raidabaugh in Journal of the American Society of Naval Engineers, Vol. XLIX, No. 4, November, 1937.

the effect of proximity of fittings on the equivalent length of a system usually has been neglected. Both these problems are considered in this paper which comprises the first of four articles to be presented on the general subject of pressure losses in marine fuel oil systems.

Fittings investigated consisted of a 3½ inch I.P.S. globe valve, angle valve, gate valve, tee, and standard and long radius elbow. In addition, three bends having ratios of radius of bend to diameter of pipe, R/d of 2, 3, and 6, and fabricated from the 3½ inch I.P.S. (150-300 p.s.i.) commercial steel connecting tubing were examined. Pertinent dimensions of the fittings are shown in Figure 1.

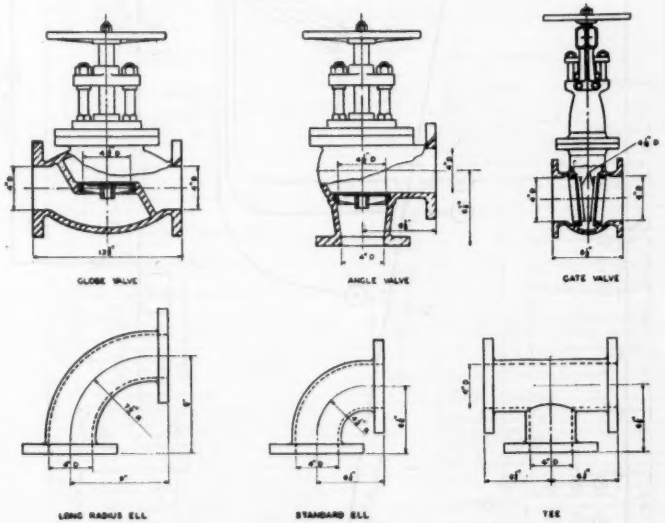


Fig.1 - FITTINGS

The set-up for determining the equivalent lengths of individual fittings (units) is shown schematically in Figure 2. For right angle fittings (bends, angle valve, etc.) the downstream section of the line was swung through a 90 degree angle. Fuel oil pumped at the required rate and temperature gave a Reynolds number range of 50 to 1000.

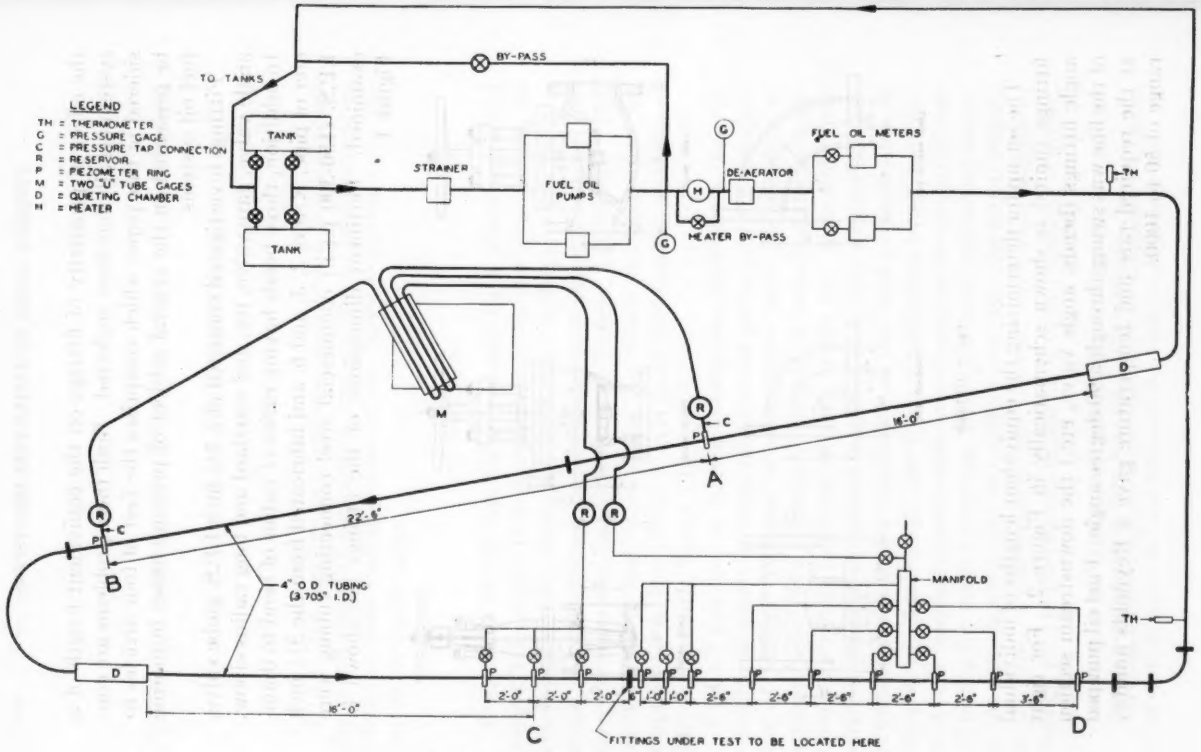


Fig 2 - DIAGRAM OF TEST SET-UP FOR INDIVIDUAL FITTINGS

ITEM NO.	ITEM	UNITS	FORMULA
1	Run Number		
2	Date		
3	Pipe Diameter		
a	Nominal		
b	Actual I.D.	Inches	Micrometer
c	Actual I.D.	Feet	Item 3b ÷ 12
4	Length of Straight Pipe		
a	Entrance length	Feet	
b	Exit length after the fitting	Feet	
c	Exit length after the fitting in diameters		
d	Piezometer ring number		
e	Length of straight pipe	Feet	
5	Oil Constants		
a	Fuel oil number		Naval Boiler and Turbine Laboratory Designation
b	Average inlet temperature	°F.	Calibrated Thermometer
c	Average outlet temperature	°F.	Calibrated Thermometer
d	Average temperature in system	°F.	(Item 5b + Item 5c) ÷ 2
e	Viscosity - Saybolt	Centistokes	Pensake Viscosimeter
f	Viscosity - English units	Sq.ft./Sec.	1.076×10^{-5} x Item 5e
g	Specific gravity 60°F./60°F.		Westphal balance
h	Specific gravity (based on 60°F. water)		Items 5g, 5d and curve
6	Manometer Constants		
a	Manometer fluid		
b	Density	g/cc	Westphal Balance
	at	°F.	Thermometer
a	Density	g/cc	Westphal Balance
	at	°F.	Thermometer
d	Connecting fluid		
e	Density at	g/cc	
f	Density at	g/cc	
g	Manometer tube area	sq.in.	
h	Reservoir area	sq.in.	
i	Manometer angle from horizontal	degrees	
j	Average manometer fluid temperature	°F.	Calibrated Thermometer
7	Oil Flow		
a	Average total flow for run	gallons	Calibrated Meter
b	Duration	minutes	Stop Watch
c	Average uncorrected rate	g.p.m.	Item 7a ÷ Item 7b
d	Meter constant		Item 7c and meter curve
e	Average corrected rate	g.p.m.	Item 7c x Item 7d
f	Average velocity	ft./Sec.	0.4085 x Item 7e ÷ (Item 3b)^2
8	Reynolds Number		Item 3c x Item 7f ÷ Item 5f
9	Manometer Readings		
	Flow Readings - Entrance Section		
a	Leg A	Inches	
b	Leg B	Inches	
	No Flow Readings - Entrance Section		
c	Leg A	Inches	
d	Leg B	Inches	
	Flow Readings - Exit Section		
e	Leg C	Inches	
f	Leg D	Inches	
g	No Flow Readings - Exit Section		
g	Leg C	Inches	
h	Leg D	Inches	
i	Pressure loss indication for entrance section	Inches	(Item 9a + Item 9b) - (Item 9c + Item 9d)
j	Pressure loss indication for exit section	Inches	(Item 9e + Item 9f) - (Item 9g + Item 9h)
10	Manometer Factors		
a	Difference in density of manometer and connecting fluids	g/cc	Items 6b, 6e, 6e, 6f, 6j, curve
b	Sine of manometer angle		Tables
c	Density factor		Item 10a x Item 10b
d	Difference in specific gravity of connecting fluid and fuel oil 60°F./60°F.		Item 5g and Items 6e, 6f
e	Reservoir level change factor		Item 6g ÷ Item 6h
f	Correction to density factor		Item 10d x Item 10e
g	Corrected density factor		Item 10c + Item 10f
h	Multiplying factor for pressure		Item 10g ÷ 27.68
11	Equivalent Length of Fitting		
a	Pressure loss indication per foot of straight pipe (entrance section)		Item 9i ÷ 22.5
b	Pressure loss indication for straight pipe in section with fitting		Item 11a x Item 4a
c	Pressure loss indication for fitting		Item 9i - Item 11b
d	Equivalent length in feet		Item 11c ÷ Item 11a
e	Equivalent length in diameters		Item 11d ÷ Item 3c
12	Pressure Losses	Lbs./Sq.in.	
a	For straight pipe per foot (from entrance section)		Item 10h x Item 11a
b	For section with fitting		Item 10h x Item 9i
c	For straight pipe in section with fitting		Item 10h x Item 11b
d	For fitting		Item 12b - Item 12a
13	Friction Factor - Average		Item 3b x Item 12a 0.387 (Item 3b)(Item 7f)^2

FIGURE 3 TEST CODE OF CALCULATIONS FOR INDIVIDUAL FITTINGS

Piezometer rings were placed at various distances downstream from the fitting under examination. The pressure drop for calculation of the equivalent length was measured from a fixed point upstream from the fitting to various points downstream and for constant Reynolds numbers. The pressure loss per foot and per diameter was obtained from a 22.5 foot section of straight tubing.

The pressure measuring system consisted of piezometer rings connected by an oil-filled line to a pressure reservoir. The top of the reservoir was filled with oil and the bottom with water. The water side of the reservoir was connected to the manometer containing colored carbon tetrachloride as the measuring fluid. Accordingly, hydraulic transmission of pressures and a differential manometer constituted the pressure measuring system.

The test code for individual fittings is shown in Figure 3.

Individual Fittings.

A disturbance is created downstream when a fluid is forced through a fitting. If the downstream piezometer ring lies in the region of the propagated disturbance, the energy dissipation will not be complete and the indicated equivalent length will not represent the total energy loss caused by the presence of the fitting. Depending upon Reynolds number, a considerable portion of the energy dissipation may take place in the tubing downstream from the fitting.

The equivalent length is rigorously defined from Figure 4. Points M, R, O, N, and Q are the indicated positions in the flow system. The actual pressure gradient is represented by $M_1R_2O_2N_4Q_4$.

$M_1M_2 = {}_M[\Delta p]_R^{\S}$ (Read pressure drop from M to R) due entirely to the characteristics of the flow in the straight tubing.

$R_2R_3 = {}_R[\Delta p]_O$ due to the resistance within the fitting.

$N_2N_3 = {}_O[\Delta p]_N$ if the section ON were not in the region of the propagated disturbance.

$Q_2Q_3 = {}_O[\Delta p]_Q$ if the section OQ were not in the region of the propagated disturbance.

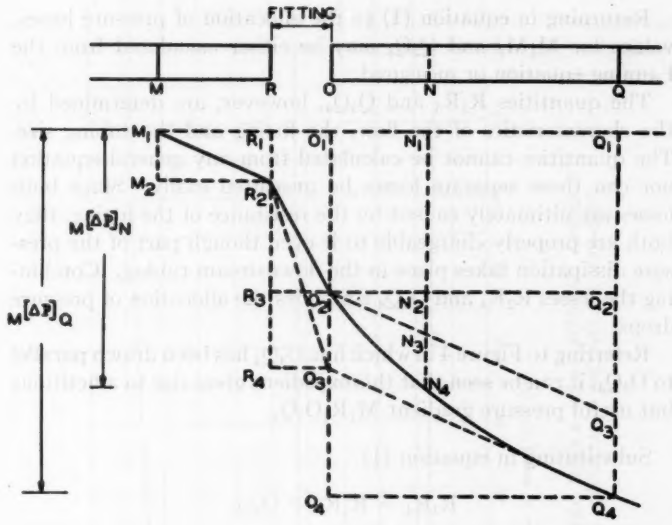


Fig. 4 - DIAGRAM OF A TYPICAL PRESSURE GRADIENT

$N_3N_4 = {}_0[\Delta p]_N$ as a result of the propagation downstream of the disturbance.

$Q_3Q_4 = {}_0'[\Delta p]_Q$ as a result of the propagation downstream of the disturbance.

Thus the total pressure drop from M to a point Q outside the region of the propagated disturbance is:

$$ {}_M[\Delta p]_Q = M_1M_2 + R_2R_3 + Q_2Q_3 + Q_3Q_4 \quad (1)$$

Correspondingly, the total pressure drop from M to a point N inside the region of the propagated disturbance is:

$$ {}_M[\Delta p]_N = M_1M_2 + R_2R_3 + N_2N_3 + N_3N_4 \quad (2)$$

Returning in equation (1) to the allocation of pressure losses, values for M_1M_2 and Q_2Q_3 may be either calculated from the Fanning equation or measured.

The quantities R_2R_3 and Q_3Q_4, however, are determined by the characteristics of the flow, the fitting and the tubing size. The quantities cannot be calculated from any general equation nor can these separate losses be measured easily. Since both losses are ultimately caused by the resistance of the fitting, they both are properly chargeable to it even though part of the pressure dissipation takes place in the downstream tubing. Combining the losses R_2R_3 and Q_3Q_4 simplifies the allocation of pressure drops.

Referring to Figure 4 in which line O_3Q_4 has been drawn parallel to O_2Q_3, it can be seen that this expedient gives rise to a fictitious but useful pressure gradient $M_1R_2O_3Q_4$.

Substituting in equation (1)

$$R_2R_4 = R_2R_3 + Q_3Q_4$$

$$O_3O_4 = Q_2Q_3$$

gives

$$_M[\Delta p]_Q = M_1M_2 + R_2R_4 + O_3O_4 \qquad (3)$$

The quantities $_M[\Delta p]_Q$, M_1M_2 and O_3O_4 can all be determined. The pressure drop chargeable to the fitting becomes

$$R_2R_4 = {}_M[\Delta p]_Q - M_1M_2 - O_3O_4 \qquad (4)$$

Equivalent length is defined as the ratio of R_2R_4 to the pressure drop per diameter of straight tubing regardless of the position of the terminal piezometer ring. If the terminal piezometer ring lies outside the region of the disturbance, as point Q in Figure 4, the equivalent length is called the real equivalent length. If it lies within the region of the disturbance, as point N in Figure 4, the equivalent length is called the pseudo-equivalent length.

Values for bend R/d = 6 are presented in Figure 5 to show the effect of the position of the downstream piezometer ring on the equivalent length. The pseudo-equivalent length becomes the real equivalent length at different distances downstream

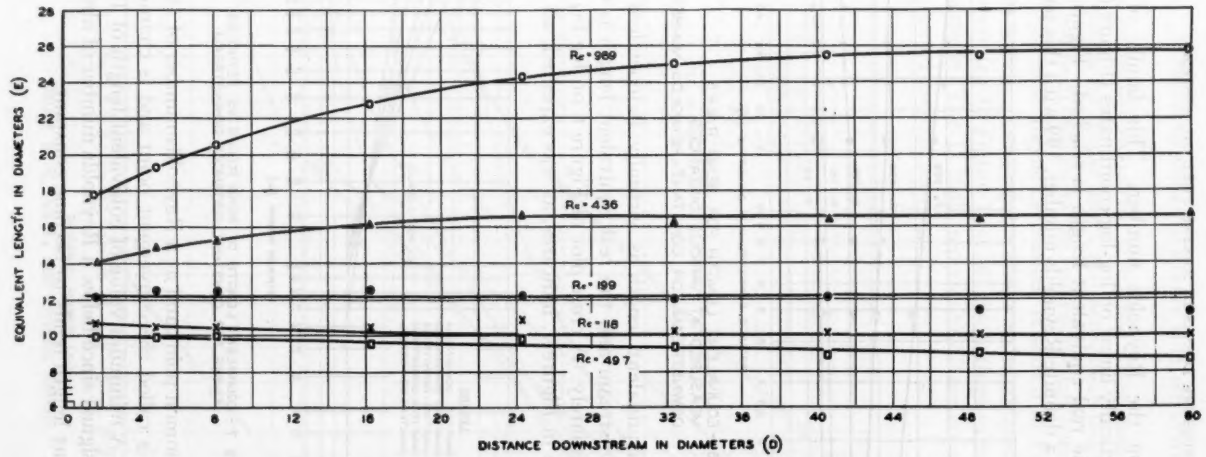


Fig. 5 - EQUIVALENT LENGTH FOR BEND R/d = 6 MEASURED AT VARIOUS DISTANCES DOWNSTREAM FOR CONSTANT REYNOLDS NUMBER

depending on the Reynolds number. This family of curves appears practically linear on log-log coordinates (Figure 6) until, of course, the real equivalent length is reached. Figure 5 indicates that at a definite Reynolds number, 199 with this particular

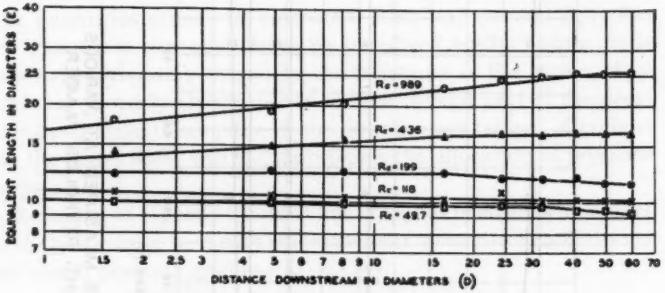


Fig. 6 - EQUIVALENT LENGTH FOR BEND R/d = 6 MEASURED AT VARIOUS DISTANCES DOWNSTREAM FOR CONSTANT REYNOLDS NUMBER

fitting, the equivalent length is virtually independent of the distance downstream and the real equivalent length is reached almost immediately. A cross plot of Figure 6 on log-log coordinates, shown in Figure 7, indicates, as was expected, all lines to

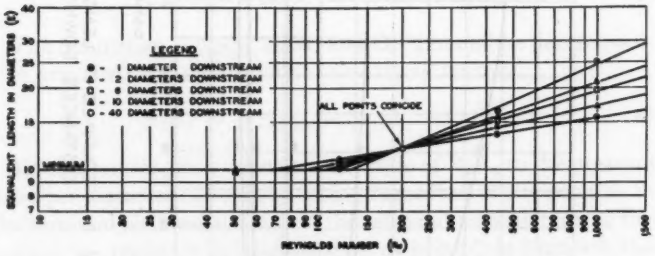


Fig. 7 - EQUIVALENT LENGTH OF BEND R/d = 6 VS. REYNOLDS NUMBER FOR CONSTANT DISTANCES DOWNSTREAM

meet at a common point and at a Reynolds number of 199. The intersection is called the convergent point and is characteristic of a fitting. A minimum value of equivalent length of 10 diameters was assigned since below a Reynolds number of about 100, the equivalent length tends to become constant.

The particular value that is reached when changes in distance downstream no longer affect the equivalent length is called E_u which has been defined as the real equivalent length. The variation of E_u with Reynolds number is shown in Figure 8 for the fittings tested. A nomogram which is the composite of the plots corresponding to Figure 7 for all the fittings examined is shown in Figure 9. The D scale represents diameters downstream of the terminal piezometer ring.

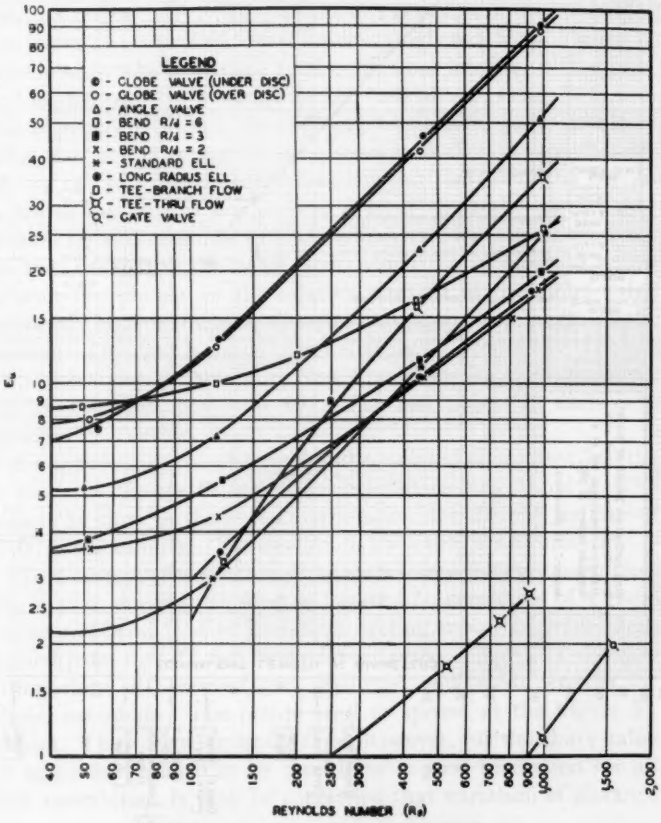

LEGEND
● - GLOBE VALVE (UNDER DISC)
○ - GLOBE VALVE (OVER DISC)
Δ - ANGLE VALVE
□ - BEND R/d = 6
■ - BEND R/d = 3
x - BEND R/d = 2
* - STANDARD ELL
● - LONG RADIUS ELL
□ - TEE - BRANCH FLOW
⋈ - TEE - THRU FLOW
Q - GATE VALVE
E_u
REYNOLDS NUMBER (R_e)

FIGURE 8. E_u VS. R_e FOR FITTINGS TESTED

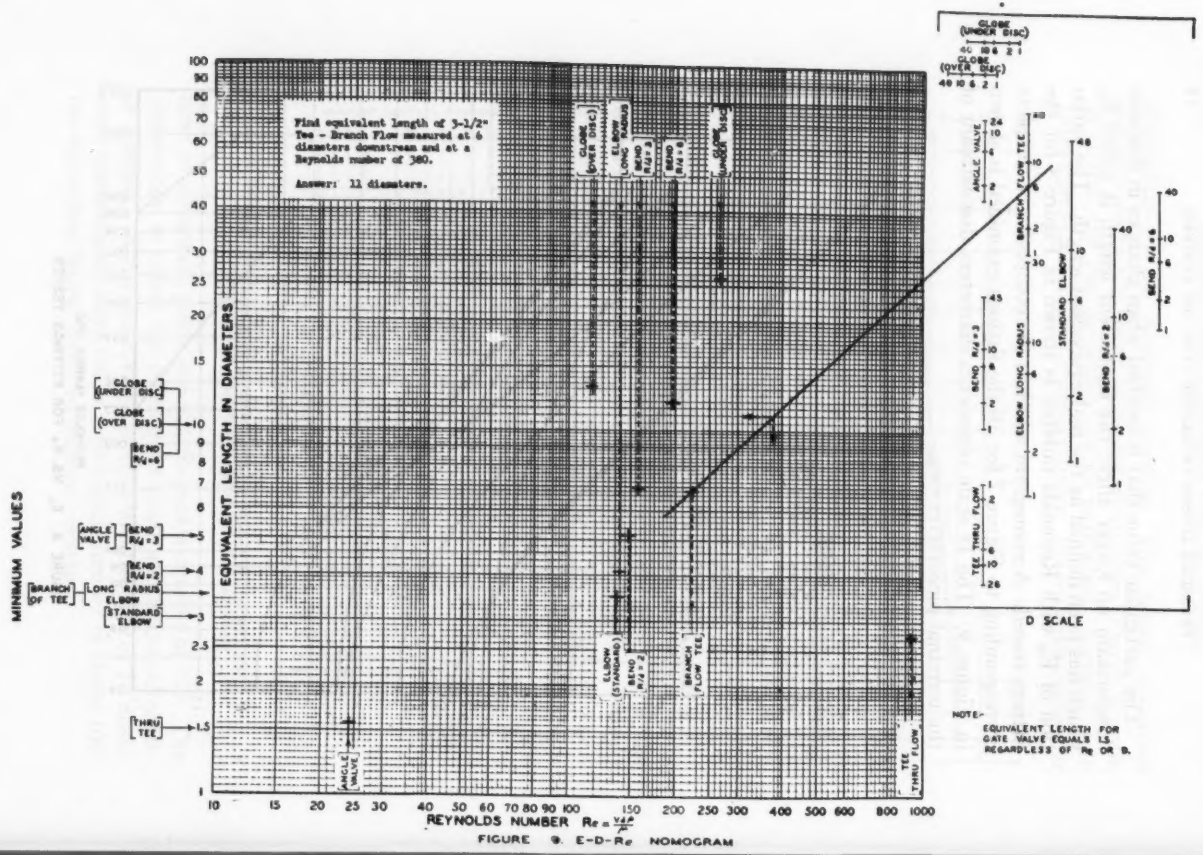

Find equivalent length of 3-1/2" Tee - Branch Flow measured at 6 diameters downstream and at a Reynolds number of 380.
Answer: 11 diameters.
EQUIVALENT LENGTH IN DIAMETERS
REYNOLDS NUMBER $Re = \frac{VdC}{\nu}$
MINIMUM VALUES
GLOBE (UNDER DISC)
GLOBE (OVER DISC)
BEND R/d = 6
ANGLE VALVE
BEND R/d = 3
BEND R/d = 2
BRANCH OF TEE
LONG RADIUS ELBOW
STANDARD ELBOW
THRU TEE
D SCALE
NOTE:- EQUIVALENT LENGTH FOR GATE VALVE EQUALS 1.5 REGARDLESS OF Re OR D.

FIGURE 9. E-D-Re NOMOGRAM

Assemblies of Fittings.

Combinations of fittings (assemblies) were investigated to determine applicability of data secured with individual fittings. The diagrams of four of the assemblies are shown in Figures 10, 11, 12, and 13.

Assemblies Nos. 1 and 2 are alike except for the position of the globe valve. It was the seventh fitting in assembly No. 1 and the third in assembly No. 2. The plots of equivalent length (exclusive of straight pipe) in diameters against feet of connecting tubing from the first piezometer ring, are indicated in Figures 14 and 15, respectively. The position of the dashed line on the right indicates the so-called chart values predicted from Figure 8, obtained by the E_u values (real equivalent lengths) of the individual fittings. In the case of the last fitting, bend $R/d = 3$, the Nomogram (Figure 9) was employed since the last piezometer ring was in the region of the propagated disturbance thereby giving rise to a pseudo-equivalent length. The use of the Nomogram in this case is not strictly correct since the disturbance caused by the assembly as a whole may not have been the same as that caused by the bend alone which the Nomogram represents. The change in the relative position of the globe valve evidently has no effect on the resultant equivalent length of the system. This fact is brought out sharply by Figures 16, 17, and 18 which compare assemblies Nos. 1 and 2 for Reynolds numbers of 50, 120, 445, 610, and 995. The last points in each case practically coincide.

Comparison of the observed and predicted equivalent lengths is made in Figures 19 and 20 for assemblies Nos. 1 and 2. The points lie close to the perfect correlation line thereby indicating good predictability. The deviation, however, is not uniform.

To determine the effect of distance between fittings, assemblies 3a, b, c, d, and e (identified in Figure 12), consisting of a globe valve and bend $R/d = 3$ with connecting tubing between them varied from 0 to 15 feet, were investigated. Figure 21 shows a comparison of observed and predicted equivalent lengths for these assemblies. The points tend to spread at the higher E_u values. They are sufficiently close, however, for the chart value of equivalent length to be considered a good prediction for all five assemblies. It may be concluded that variation of distance

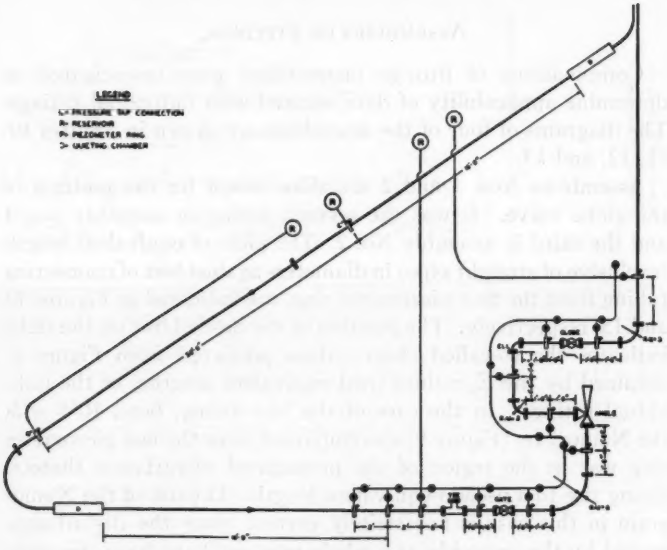


Fig 10 - DIAGRAM OF ASSEMBLY NO. 1

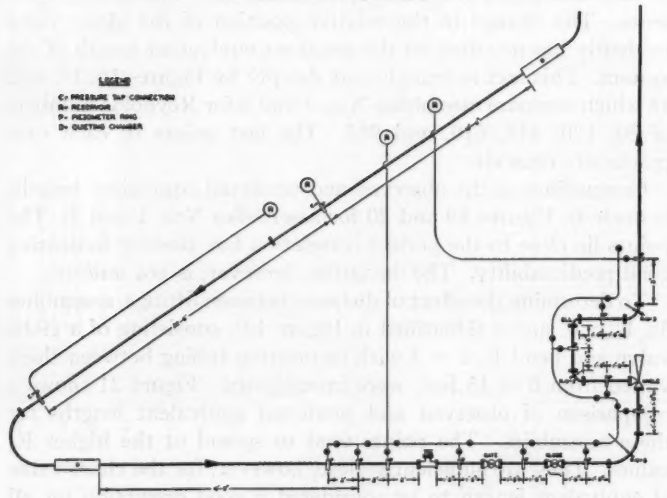


Fig 11 - DIAGRAM OF ASSEMBLY NO 2

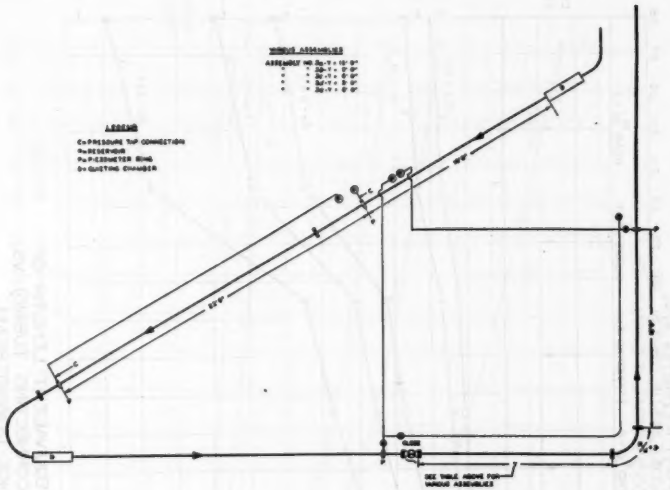


Fig 12 - DIAGRAM OF ASSEMBLY NO. 3

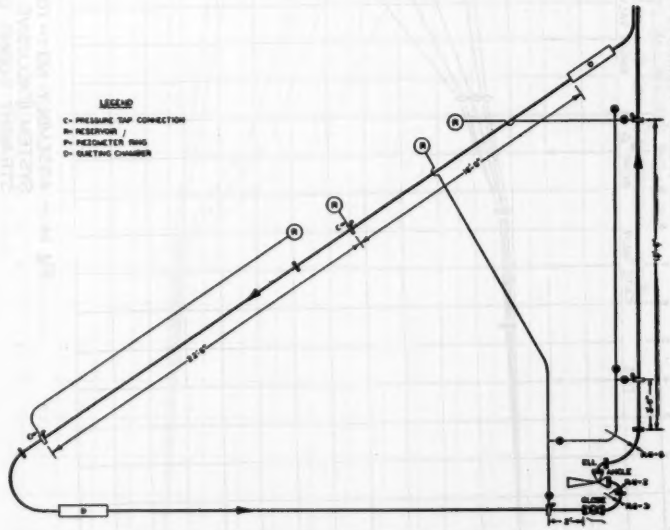


Fig 13 - DIAGRAM OF ASSEMBLY NO. 4

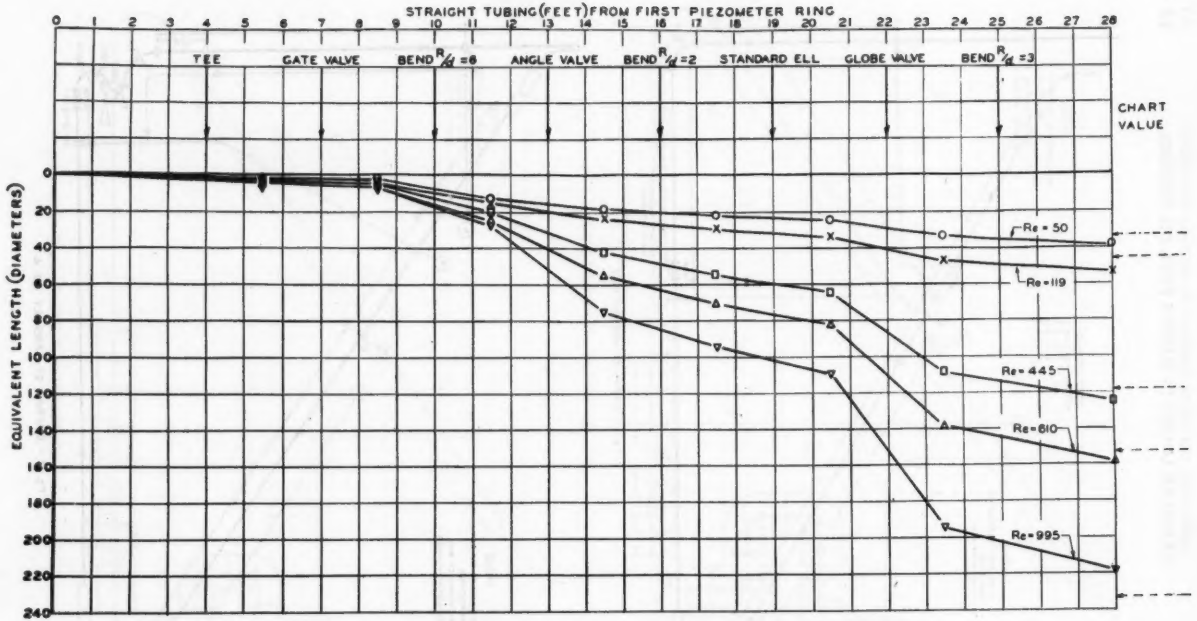

STRAIGHT TUBING (FEET) FROM FIRST PIEZOMETER RING
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28
TEE
GATE VALVE
BEND R/d = 6
ANGLE VALVE
BEND R/d = 2
STANDARD ELL
GLOBE VALVE
BEND R/d = 3
CHART VALUE
EQUIVALENT LENGTH (DIAMETERS)
0 20 40 60 80 100 120 140 160 180 200 220 240
Re = 50
Re = 119
Re = 445
Re = 810
Re = 995

Fig. 14 – ASSEMBLY NO. 1 – TOTAL EQUIVALENT LENGTH OF SYSTEM (EXCLUSIVE OF CONNECTING TUBING) VS. STRAIGHT TUBING DISTANCE DOWNSTREAM

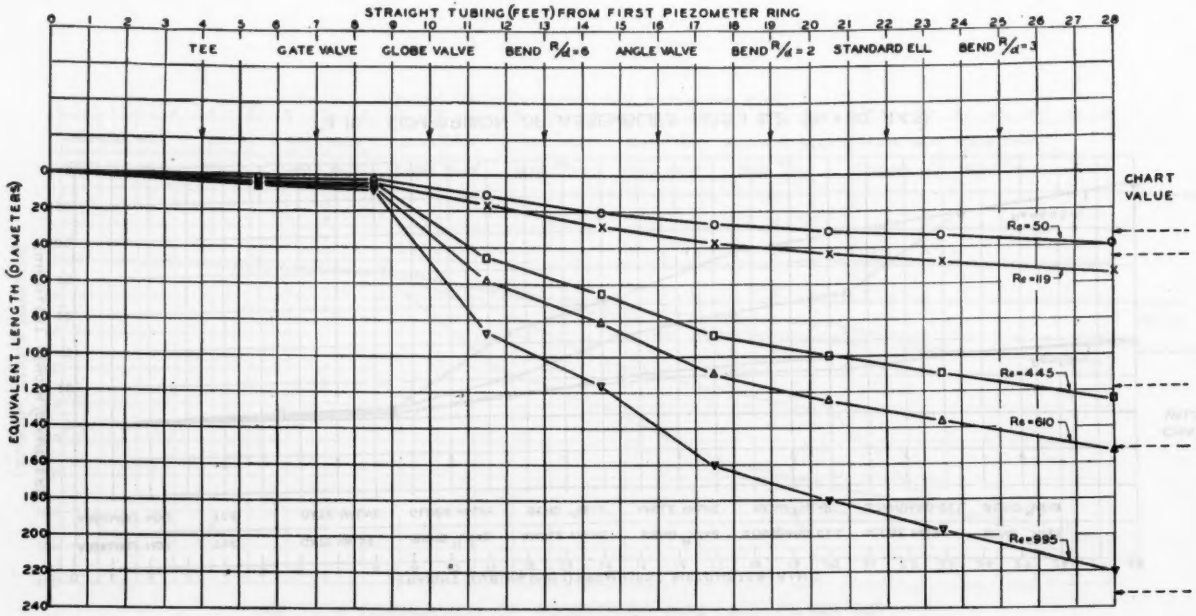

STRAIGHT TUBING (FEET) FROM FIRST PIEZOMETER RING
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28
TEE
GATE VALVE
GLOBE VALVE
BEND R/d=6
ANGLE VALVE
BEND R/d=2
STANDARD ELL
BEND R/d=1
EQUIVALENT LENGTH (DIAMETERS)
0 20 40 60 80 100 120 140 160 180 200 220 240
CHART VALUE
Re=50
Re=119
Re=445
Re=610
Re=995

Fig. 15 - ASSEMBLY NO. 2 - TOTAL EQUIVALENT LENGTH OF SYSTEM (EXCLUSIVE OF CONNECTING TUBING) VS. STRAIGHT TUBING DISTANCE DOWNSTREAM

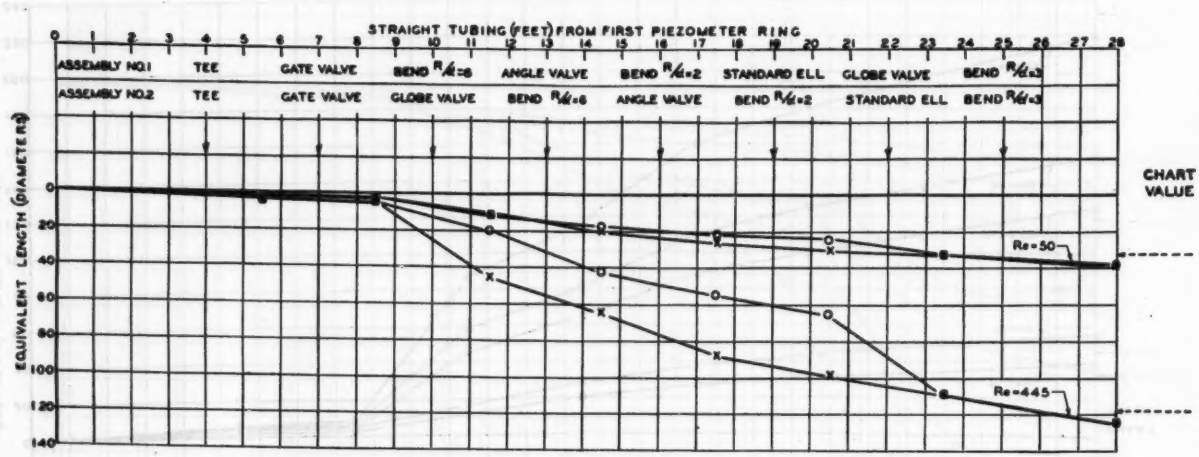


Fig. 16 - COMPARISON OF ASSEMBLIES NOS. 1 & 2 (Re = 50, 445)

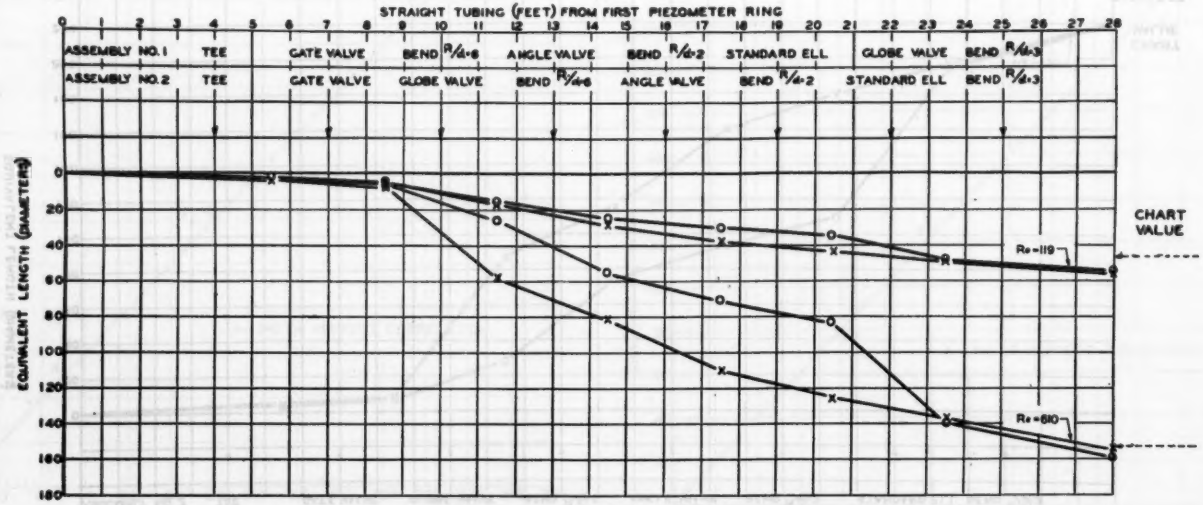


Fig. 17 - COMPARISON OF ASSEMBLIES NOS. 1 & 2 (Re=119, 610)

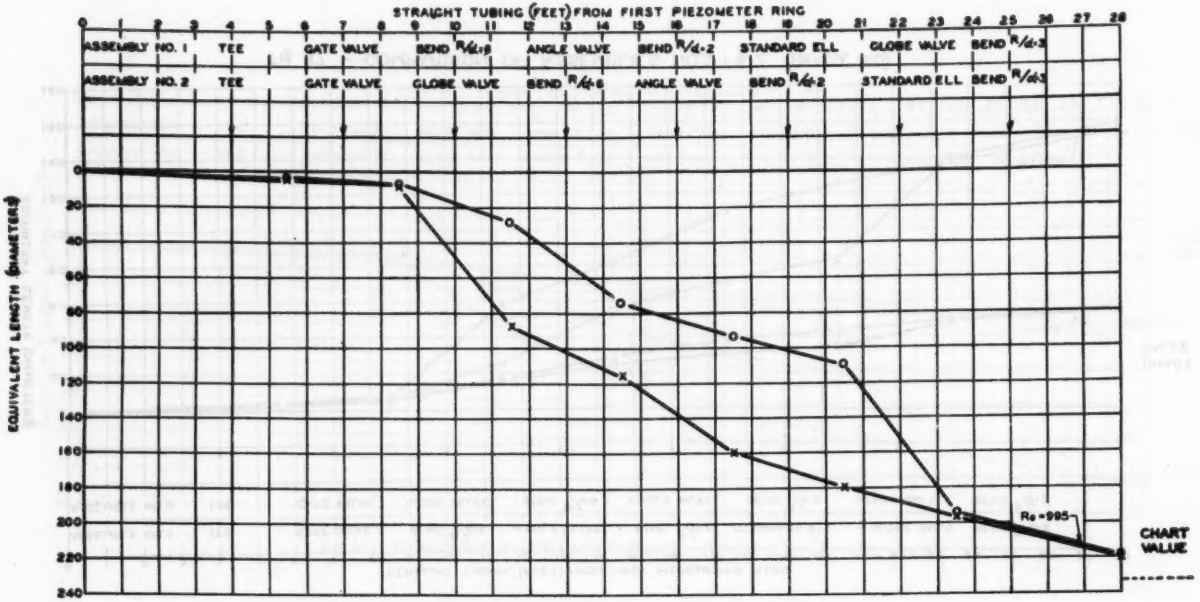


Fig. 18 - COMPARISON OF ASSEMBLIES NOS. 1 & 2 (R_e=995)

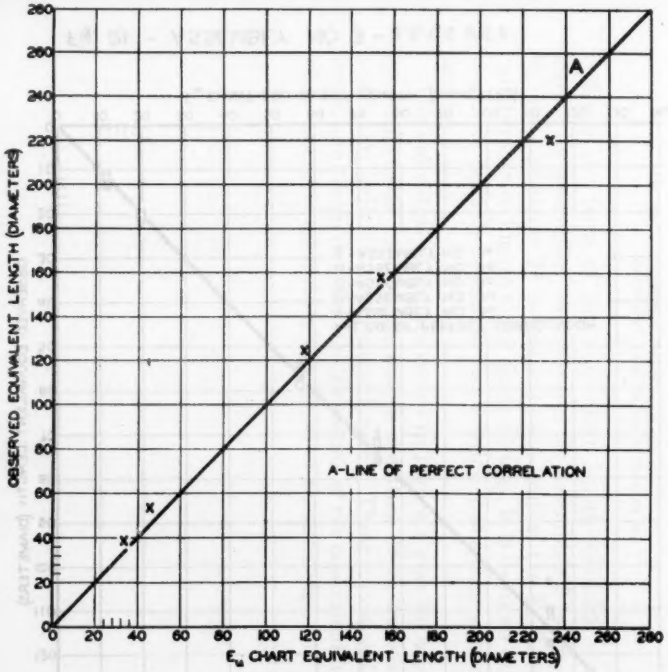


Fig. 19 - ASSEMBLY NO. 1, OBSERVED EQUIVALENT LENGTH VS. E_U CHART EQUIVALENT LENGTH

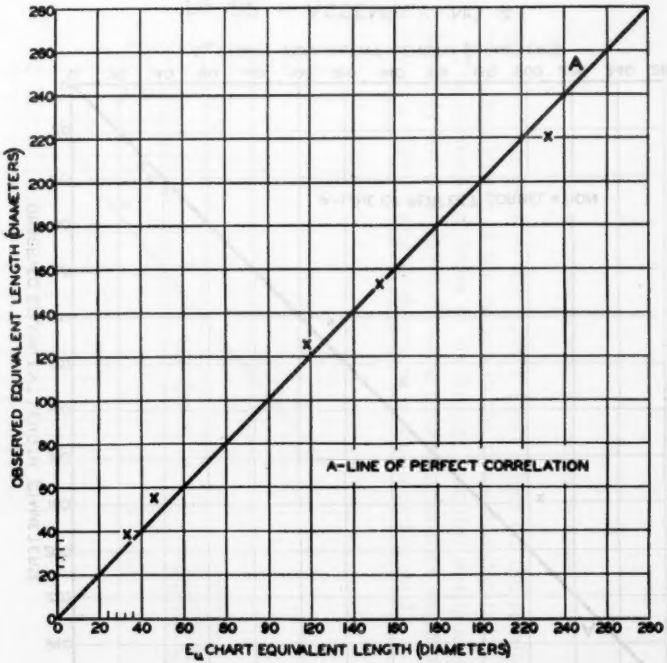


Fig. 20 - ASSEMBLY NO. 2, OBSERVED EQUIVALENT LENGTH VS. E_U CHART EQUIVALENT LENGTH

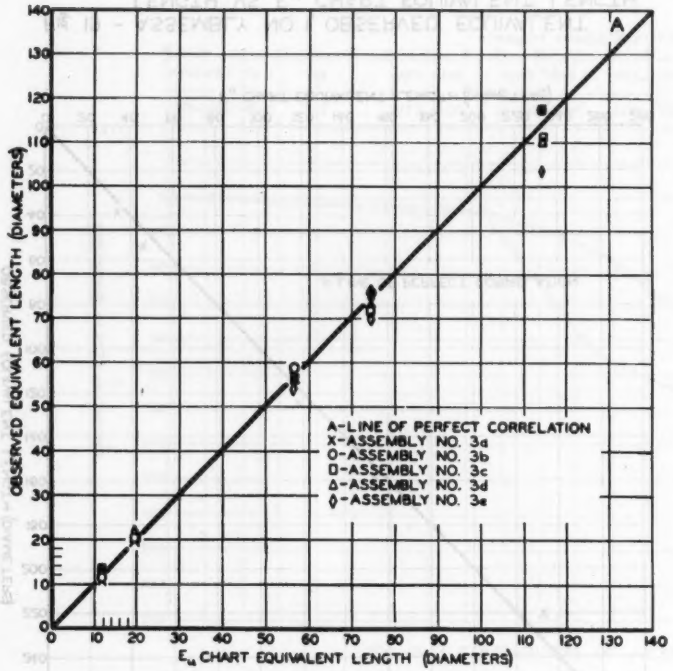


Fig. 21 - ASSEMBLY NO. 3 - a, b, c, d and e.

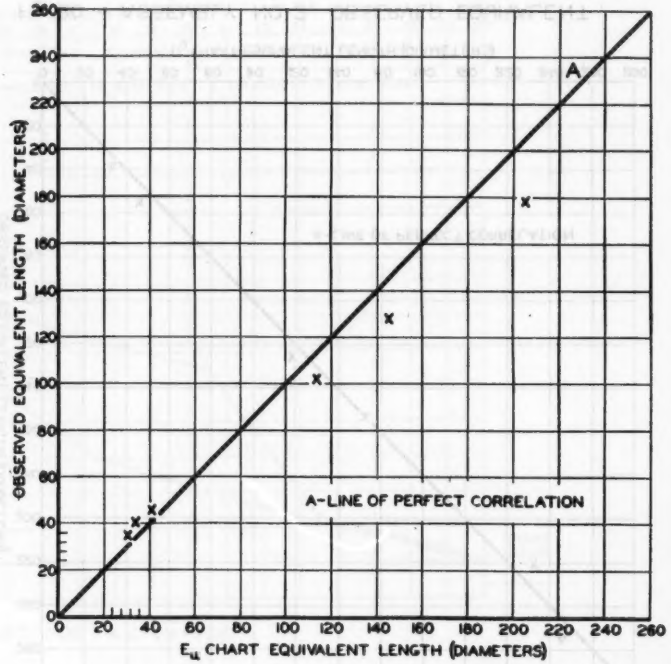


Fig. 22 - ASSEMBLY NO. 4

between fittings does not have a great effect on the resultant equivalent length of the system in the range investigated.

The extreme condition of six fittings installed contiguously, assembly No. 4, discloses the same trends as the other assemblies but to a slightly greater degree as is shown in Figure 22.

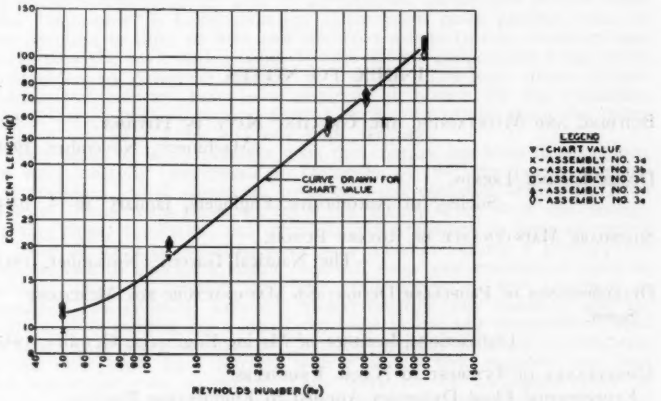


Fig. 23 - ASSEMBLY NO. 3 - a, b, c, d and e
EQUIVALENT LENGTH VS. REYNOLDS NO.

Since the form of the equivalent length-Reynolds number curves for assemblies, as illustrated for assemblies Nos. 3a, b, c, d, and e in Figure 23 is similar to the form of the E_u-Reynolds number curve (Figure 8) for single fittings, it is deduced that an assembly behaves essentially like a fitting with a large equivalent length.

The individual E_u values shown in Figure 8 may be adjusted to compensate for the deviations resulting from methods of combining individual equivalent lengths. These curves will then constitute design data for the calculation of the equivalent length of an assembly comprising any or all of these fittings.

NOTES.

INDEX TO NOTES.

Building and Maintaining the Greatest Navy in History.
—Machinery, November, 1943.

Diesel Engine Trends.
—Society of Automotive Engineers, January 10-14, 1944.

Shipboard Maintenance of Rotary Pumps.
—The Nautical Gazette, November, 1943.

Developments in Propeller Design and Manufacture for Merchant Ships.
—Transactions, Institute of Marine Engineers, October, 1943.

Miscellanea of Interest to Naval Engineers.
Experimental Fluid Dynamics Applied to Engineering Practice.
—North-East Coast Institution of Engineers and Shipbuilders, 29 October, 1943.

A Plea for the Revision of the Tonnage Laws.
—Transactions, the Institution of Engineers and Shipbuilders in Scotland, November, 1943.

Electronic Tubes—How They Work.
—General Electric Review, November, 1943.

New Supercharger for Marine Engines.
—The British Motor Ship, October, 1943.

Barge Places 90-Ton Breakwater Blocks.
—Journal, Franklin Institute, September, 1943.

BUILDING AND MAINTAINING THE GREATEST NAVY IN HISTORY.

Rear Admiral E. L. Cochrane, the Chief of the Bureau of Ships, United States Navy Department, is author of this article, reprinted from Machinery, November 1943.

Phenomenal records in ship construction and repair are being made on American shipways today, both in Government-owned and private yards. The "impossible" is being done—an achievement made possible, first, by the devotion to duty of men and women workers (many of whom have never seen the sea), and second, by the all-out cooperation being given shipbuilders by American industry as a whole. The joint efforts of individuals and industry have been inspired, of course, by the realization that a United Nations victory depends, to a great extent, upon the ability to keep our guns firing and our ships moving.

Once again, as in all previous wars this country has been dragged into, we were woefully unprepared on December 7, 1941—the Japanese gained a tremendous initial advantage by their well executed attack on Pearl Harbor. Even with assistance from our Allies, there were not enough fighting ships, auxiliaries, and other shipping facilities to hope or expect to hang on to what we then possessed. Thus it became a primary but difficult task to allocate available facilities, delay our enemies as much as possible while contracting our lines, and accept as few sacrifices as possible. The enemies, although underestimating our industry and resources, were striving for a knock-out blow before our potentiality could be felt. They were thwarted at Coral Sea, quickly followed by Midway, where they must have learned with a shock how superior our men and equipment were to theirs. They were yet by no means ready to go on the defensive, for they realized that this inevitably would lead to defeat.

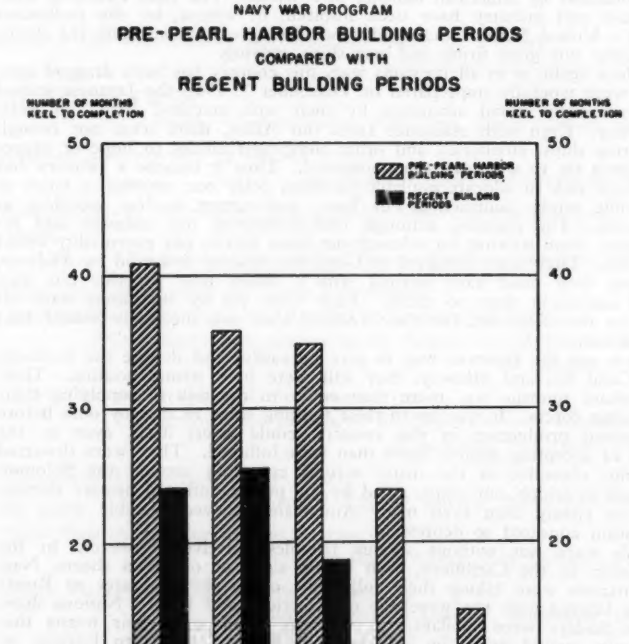
It is not the Japanese way to give up easily, and despite the set-backs at Coral Sea and Midway, they still were in a strong position. Their merchant tonnage was more than equal to the task of supplying their outlying forces. It was up to their fighting ships to destroy ours before increased production in this country could assert itself, even at the cost of accepting greater losses than were inflicted. They were thwarted in this objective in the many actions centering around the Solomon Islands in which our ships, aided by air power, inflicted heavier damage to the enemy than even many Americans believed possible when the situation appeared so depressing.

We were not without serious troubles ourselves, however. In the Atlantic, in the Caribbean, even within sight of our own shores, Nazi submarines were taking their toll. The ocean thoroughfares to Russia were littered with the wreckage of American and United Nations ships. Axis leaders were jubilant; the air was filled with their boasts that America would never be able to send invasion forces to Europe, or even to keep England from starving. Obviously and admittedly, herculean efforts were required to meet the situation. How successful these efforts were, is best born out by examining the record.

But first, let us take a look at the problem. The primary function of our Naval Shore Establishment is to provide service to the Fleet. The Navy Yards perform a vital part of this service in keeping our fighting ships fit and ready for service, providing them with the latest types of fighting equipment, and healing their wounds after battle. Today, not only are the yards doing notable work in building new ships, but they must be and are ready at all times to care for the ships on the fighting

line. In times of peace this maintenance function is fulfilled by the assignment of ships to home yards, where they return at regular intervals for well defined overhaul periods which are assigned many months in advance. On this basis, the work at the yards can be scheduled in an orderly manner and a fairly constant working force employed on this type of work. The larger Navy Yards in peacetime ordinarily employ from 5000 to 7000 men, working forty hours a week with a little shift or over-time work.

After Pearl Harbor an entirely new situation had to be met. All ships had to be brought to their highest state of efficiency and maintained there. New developments in weapons and instruments had to be rushed through production and be installed. Extended cruising in convoy



and other work placed a new emphasis on maintenance, but the time allowed for this in the yards was sharply curtailed. Then came the battle damage repair jobs which had not and could not be scheduled.

To meet this situation, our Navy Yard working forces were doubled, tripled, and quadrupled, as fast as men could be recruited. When skilled men were no longer available, untrained men and women were brought in to work alongside the skilled until they could pick up their share of the work load. As an example, the force at one of our East Coast Navy Yards expanded from 1500 men in 1937 to 26,000 in 1943. One of our larger Navy Yards now employs over 69,000 men and the forces are still increasing in some areas.

Getting the ships damaged at Pearl Harbor back into service was a large order. Workers were recruited, at first largely from the mainland Navy Yards, for this task. Pearl Harbor Yard grew, like a mushroom, almost overnight to many times its former size. The workers were subjected to inadequate accommodations, and were separated from their families and friends, but ungrudgingly accepted these conditions. The Japanese can testify as to the effectiveness of their work.

As the size of the Navy increases, this repair load grows in direct proportion. On December 1, 1941, just before Pearl Harbor, there were less than a thousand ships in commission in the Navy. On July 1, 1943, this number was approaching four thousand, excluding landing craft. During one month of this year, there were 436 naval vessels under repair in nine Navy Yards; work costing a total of $20,000,000 was accomplished on these ships. Most of this repair work is done on the basis of returning the ships to the Fleet at the earliest practicable date; to accomplish this, full-shift work and over-time is employed. In many cases where important units of the fleet were needed by specified dates, the yard personnel have virtually accomplished the impossible in meeting the deadline. There are many unsung heroes among our shipyard workmen and foremen who have worked long hard hours without benefit of blueprints to get a ship back to the battle line in record time.

Some notable jobs have been done in repairing ships that were badly damaged in battle. Well known to the public are the cases of the *Shaw* and the *San Francisco*. The former, which lost everything forward of her No. 1 stack at Pearl Harbor, was fitted with a false bow and steamed to Mare Island under her own power, where a new bow was waiting in drydock. The false bow was removed and the new one installed, and all other repairs completed in less than four months. The *San Francisco* was restored after her battering in the battle of Guadalcanal in a period of two months, a better ship than when she entered the battle.

As the repair load began to grow by leaps and bounds, it became evident that our Navy Yards would not be able to handle all of the work on naval vessels, and plans were made to make use of private ship repair yards. There were many difficulties to overcome in setting up this program, due to the unfamiliarity of the private yards with Navy specifications and standards, but they were overcome by the hearty cooperation of the shipyard personnel, Naval inspectors and ships' officers. Most of these yards have now passed the educational stage and take Navy work in their stride. The extent of this program is indicated by the fact that during one month of this year there were 418 U. S. and foreign naval vessels undergoing overhaul or conversion in private yards in this country.

Actually a sizable part of the work undertaken in our yards, both Government and private, has been the overhaul and repair of foreign warships. During the past year this work constituted 17 per cent of the total repair load, and involved an expenditure of approximately $60,000,000. In all, eighty-three large ships were repaired for ten Allied nations, the majority being British. This work, especially in the beginning, taxed the

resourcefulness of our yards in adapting U. S. methods and materials to the requirements and specifications of foreign naval services. The use of different names for the same article by the U. S. and British Navies did not help the situation. Our workmen had to learn that an elevator on a British ship is a "lift," that a wire is "earthed" rather than grounded, that a "bird cage" is a radar shack, and that a "wandering lead" is a portable cable.

Many unusual problems have arisen during the repair of damaged ships for which there was no precedent, and the yards have displayed unusual resourcefulness and ingenuity in solving these. As an example, a ship arrived at a West Coast Navy Yard for repair of damage caused by an explosion in one of her gasoline tanks. The deck plating had been forced downward in such a way as to block access to the damaged gasoline piping. It had not been practicable to remove all gasoline in the area, and some preliminary cutting away was necessary before steaming of the tanks could be undertaken. The yard was therefore faced with the problem of cutting away the plating in an explosive atmosphere.

The problem was solved by flooding the compartment with water, and sending down a diver to cut away the plating with underwater cutting equipment. The cut plating was then removed and the water pumped down. As the arc type of underwater cutter was not available, the gas type was used. This necessitated lighting the torch on the forecastle deck and lowering it to the diver in a bucket of water. As a further precaution, the bucket and torch were kept in a fog spray until the torch was submerged. Access was then had to the damaged piping. During the work of removing the piping, it became necessary to do some chipping. This was accomplished by putting a chipper in rain clothes, and he accomplished the necessary chipping enveloped in a fog spray from a conventional fog nozzle. Having completed this preliminary clearing away, the steaming and gas freeing of the tanks proceeded.

A cruiser had her bow blown off by torpedoes to a distance of 80 feet from the stem. While the ship was undergoing temporary repairs to permit her to reach a Navy Yard, work was started on a new bow in the yard shops. Twenty-five days later when the ship arrived at the yard, a new bow section had been completely assembled and set up in drydock. Within two weeks of her arrival the new bow was installed and the ship could be undocked.

The new bow was constructed in the shop in three units, by dividing the bow into three horizontal layers. Each of these were later cut into two sections to permit handling by yard cranes. The upper units were built in an inverted position to permit "down welding" of beams and bulkheads to decks.

Operations of our ships under war-time conditions have resulted in some cases of severe damage from collision. A modern destroyer was cut almost in half by another vessel, the bow penetrating between the two boilers to within six feet of the center line. The keel was knuckled and the forward part out of line with the after part to the amount of 3½ feet at the bow. The stern section was also twisted relative to the bow section. Attempts to straighten the ship when resting lightly in the docking blocks failed, and it was decided to cut the ship completely through, realign the sections and join them again. As the forward section was unstable by itself, this was ballasted and strapped to the dock floor. The after section was then floated, aligned, and listed to correspond with the forward section, and then landed on the blocks. When the repairs had been completed, the work had been so skillfully accomplished that it was not necessary to re-machine the gun foundations, as these were within tolerable limits of parallelism.

Recently one of the Navy's cargo ships sustained serious underwater

battle damage caused by a torpedo explosion. Repairs were accomplished by a Navy Yard in a remarkably short time, due not only to the efficient work of the yard, but also to the excellent preliminary damage surveys and work lists forwarded by the vessel from overseas only a few weeks after the occurrence of the casualty. As a result of the submission of the preliminary and revised work lists of necessary repairs and material, it was possible for the yard to order and have on hand, upon the arrival of the vessel, more than a million dollars worth of material consisting, in part, of 900 tons of structural steel. This made it possible to drydock the ship immediately upon her arrival and accomplish all repairs within two months after that date.

Up to the start of the war, lessons learned from battle damage to combatant ships whenever obtainable were carefully analyzed and made as applicable as possible in current construction and in alterations to ships of our operating fleet. In order to increase production rates, and to take care of the design of new classes of craft (which experience was indicating daily would be required) design of ships was frozen, in so far as practicable, so that production-line construction could be realized. It was necessary for important changes to be squeezed into production lines as unobtrusively as possible. Vast improvements in industrial knowledge and products were being realized under the impetus of war. In the design of a warship, a compromise between space and weight requirements for speed, protection, and armament must be struck. Whenever improvements in any of these could be incorporated in construction at a saving of weight and space, the "payload" or offensive power could be increased accordingly. Although there was no over-all saving in weight, there has been a vast improvement in the effectiveness of its use.

The goals set up by the President and believed by so many to be impossible of accomplishment have been met or exceeded. Men and women from all walks of life and from every town and city in the country have flocked to shipbuilding and repair centers. Living conditions were at times almost unbearable and the unaccustomed work tiring, but the majority stayed and made possible the situation as it is today. Women have contributed in large measure to the program. They are daily taking on new work in shipyards or replacing men leaving to join the Armed Forces.

In spite of freezing the design of ships to a large extent, improvements are finding their way into the final product at an astounding rate. This is brought about in many ways. The National Defense Research Committee and many other Government agencies are contributing to a marked degree. The workers themselves are thinking of better and faster ways of accomplishing the desired ends. Although, in most cases, the worker submits his ideas with the sole objective of improving the war effort, monetary rewards and suitable recognition of merit are made. It appears to those whose task it is to weed out the practical from the impractical, that the nation has turned into a land of inventors, so enthusiastic are those engaged in shipbuilding in their efforts to contribute.

The most impressive and the most satisfying statistics as regards shipbuilding are the reduction in man-hours required to turn out a ship of a given class, although the ships themselves are increasingly complicated. This reduction has been in round figures, 30 per cent since our entry into the war. This has been brought about by various causes, but the primary one is the effort, enthusiasm, and the "will to win" of the workers.

Building periods have been shortened greatly through the increased experience and knowledge of the personnel, improvements in methods, and the number of personnel actually working at the task. As examples of the improvements in building periods of ships realized since our entry into the war, the following are given:

Type	Building Period Reduced, Per Cent
Heavy Cruisers	41
Light Cruisers	29
Large Aircraft Carriers	47
Destroyers	69
Submarines	49

An impressive comparison is that of the main propulsion horsepower installed in completed combatant ships on December 7, 1941, and at present.

On December 7, 1941, about 13,000,000 H.P.
On September 1, 1943, about 32,000,000 H.P.

The contributions and the efforts of the millions engaged in this mighty effort to defeat our enemies in the shortest possible time and at the cost of as few lives as will suffice are not lost to posterity. The improvements in methods, the more effective employment of materials, the substitution of one material for another as one becomes more critical is knowledge gained which can be applied to a better way of life when the task on hand is accomplished.

DIESEL ENGINE TRENDS.

This is an abstract of a paper read by Edward C. Magdeburger before the Annual Meeting of the Society of Automotive Engineers at Detroit, Michigan, January 10-14, 1944. Mr. Magdeburger is Head Engineer of the Diesel Section of the Bureau of Ships, Navy Department. He gives an interim report of the considerable Diesel engine experience being gathered by the United States Navy at war, particularly design modifications already shown to be very desirable.

The purpose of this paper is to take advantage of the unusual circumstances created by the war, whereby practically the whole Diesel engine industry of this country contributed not only great numbers of the standard types of engines but also developed many new and improved designs to meet the Navy's war needs. It is therefore intended to present the design characteristics of the Diesel engines used in the Navy and point out some of the probable design trends.

The principal characteristics of the engines for the Navy are light weight, compactness and reliability of performance. The justification of these requirements can be easily visualized in the light of everyday war news. These are the requirements which will help American engines to survive the competition of the post-war world.

Exact figures on number of engines or their total output are not essential for our discussion. Such figures, however, were used in evolving the picture it is intended to present. Figure 1 gives the percentage of the total horsepower creditable to every cylinder size. Since most of the sizes had less than 1 per cent of the total, two different scales of ordinates were used. The upper curves in this figure give the Rpm. and horsepower output of an individual piston, for both 4 cycle and 2 cycle engines at full power rating, based on 20 of the ranking engines, the data for which is assembled in Table I. These curves represent actual data for engines in greatest use in the Navy and not any one's individual opinion as to what the present day Diesel engine is or should be.

The most striking observation from the study of Figure 1 is the fact that 47 per cent of all Diesel engines in the Navy are of the 4¼-inch

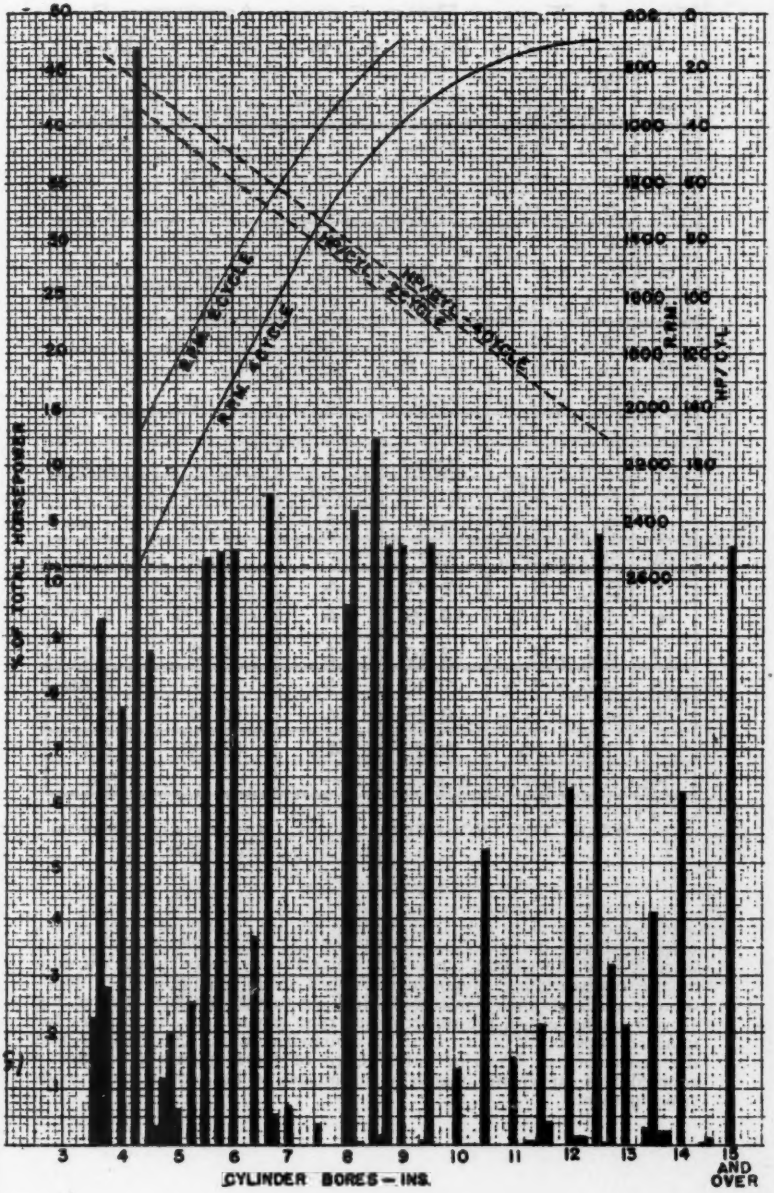

% OF TOTAL HORSEPOWER
R.P.M.
H.P./CYL.
4 STR. 4 CYCLE
2 STR. 4 CYCLE
2 STR. 2 CYCLE
CYLINDER BORES — INS.
3
4
5
6
7
8
9
10
11
12
13
14
15 AND OVER

FIGURE 1.

TABLE I. TWENTY DIESEL ENGINES ARRANGED BY PERCENTAGE OF TOTAL HORSEPOWER IN U.S. NAVY.

No.	Bore & Stroke	R.P.M.	Cycle	B.H.P. Per Piston	% Total B.H.P.
1	4 1/4 x 5	2100	2	33.3	38.8
2	6 1/2 x 7	1270	2	62.5	7.02
3	8 1/2 x 10	744	2	75.0	6.33
4	8 1/8 x 10	800	2	85.0	5.55
5	4 1/4 x 5	2100	2	33.3	4.59
6	8 1/2 x 10 1/2	750	2	100	2.68
7	9 1/2 x 12	900	4s x)	112.5	2.57
8	6 x 6 1/2	1620	2	60.0	2.48
9	8 1/2 x 10 1/2	750	2	85.0	2.36
10	8 3/4 x 10 1/2	750	2	106.3	2.38
11	5 3/4 x 6	2000	4s	41.7	2.31
12	4 1/4 x 4 3/4	2600	4s	25.0	1.71
13	9 1/16 x 13 3/8	700	2	100.0 xx)	1.64
14	9 1/16 x 13 3/8	700	2	85.4	1.16
15	5 1/2 x 7	1200	4	18.75	1.13
16	3 5/8 x 4 1/2	1425	4	6.25	0.87
17	4 1/4 x 5	1200	2	15.0	0.81
18	4 1/4 x 5 1/2	1700	4	17.5	0.80
19	4 x 5 1/2	1885	4	10.0	0.75
20	12 1/2 x 13	700	4s	150.0	0.71
ALL OTHERS		2 & 4	—	—	13.4

x) 4s = SUPERCHARGED xx) SEPARATELY DRIVEN BLOWER

TABLE I.

bore which develops 33 1/3 horsepower per piston at 2100 Rpm., when operating on a 2-stroke cycle and 25 horsepower at 2600 Rpm. when it is of the 4-cycle type. It is not necessary to call your attention to the overwhelming preponderance of 2 cycle engines and of high-speed engines of small bores in Table I, or to the fact that the ranking 4 cycle engines are of the supercharged type leaving only the engines for standard ships' boats in the naturally aspirated class and that only because the desirability of "freezing" the design of these engines as originally adopted was far greater than any advantages to be derived from modernizing them.

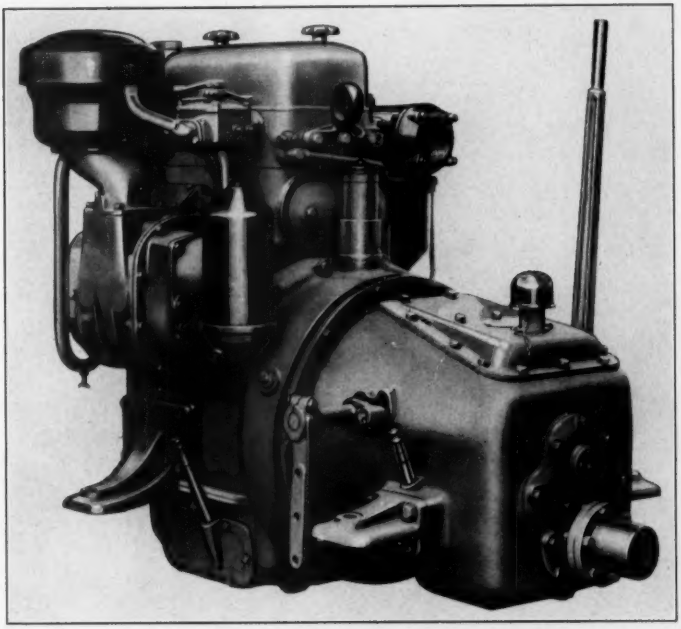


FIGURE 2.

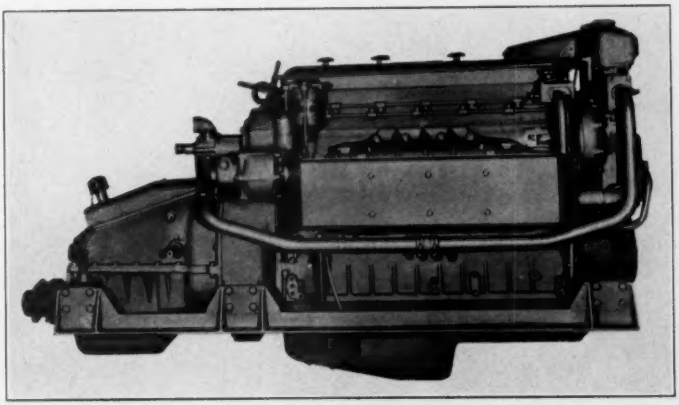


FIGURE 3.

FIGURE 4.

Craftsmanship vs. Mass-production. With the development of mass-production methods and the adaptation of the American automobile to the needs of the masses, the number of automobile owners in this country grew into millions. Compare this with the hand-made perfection of the European automobile. In a similar manner, the Diesel engine never reached its maxim usefulness until adopted by the automotive industry under pressure of war and produced on assembly lines with greatly increased speed and with consequent reduction in price.

This process of adapting the Diesel engine to mass-production robbed it of its aura of craftsmanship, which surrounded the process of assembling and testing a Diesel engine only a few years ago. But it gained immensely in adaptability. This new adaptability of the Diesel engine is best illustrated by the ranking engine of Table I and is manifestly the reason for its magnificent usefulness in this war. This cylinder size is used in the Navy in engines from 16 to 800 horsepower and in units of from one to 24 cylinders. Figures 2, 3 and 4 illustrate the single-cylinder, 6-cylinder, and 24-cylinder engine units of this size. The second engine of Table I has likewise a notable range of power—from a 3-cylinder engine driving a 100 Kw. generator unit to a 32-cylinder engine of over 2000 horsepower output.

Thus the advantages of mass-production forced the development of units of greater number of cylinders since only by so doing could the Navy obtain the desired ranges of engine output.

Cylinder Arrangement then became the problem and I would like to illustrate diagrammatically all of the arrangements used to date. These have all been drawn to the same scale around a cylinder of the same bore and stroke and illustrate at a glance the space requirements of each arrangement.

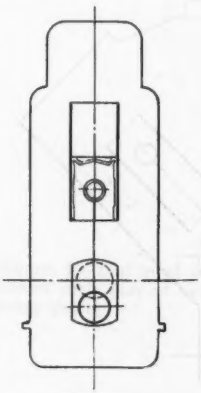


FIGURE 5.

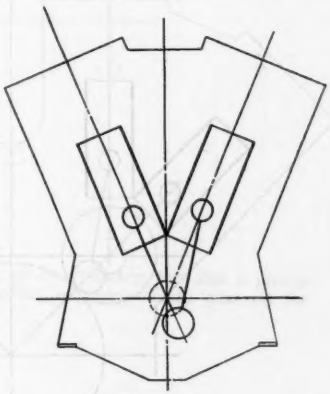


FIGURE 6.

Figure 5.—The basic 2-cycle cylinder with unaflow scavenging through exhaust valves in the cylinder head.

Figure 6.—The standard V arrangement of two cylinders, built in 6, 8, 12 and 16 cylinders, both with side-by-side connecting rods and with fork-and-blade type of rods.

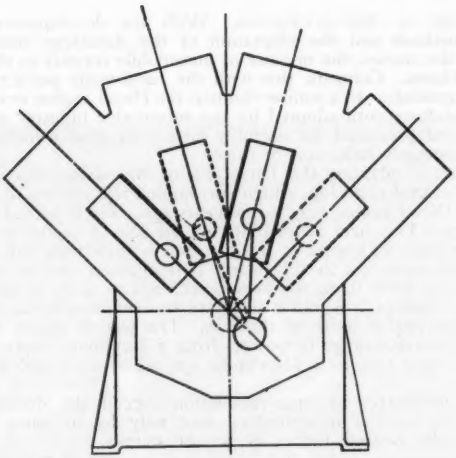


FIGURE 7.

Figure 7.—The porcupine type with four cylinders or two Vees arranged on a single crankpin and two pairs of fork-and-blade rods arranged side by side. Used in producing a 32-cylinder unit on one crankshaft.

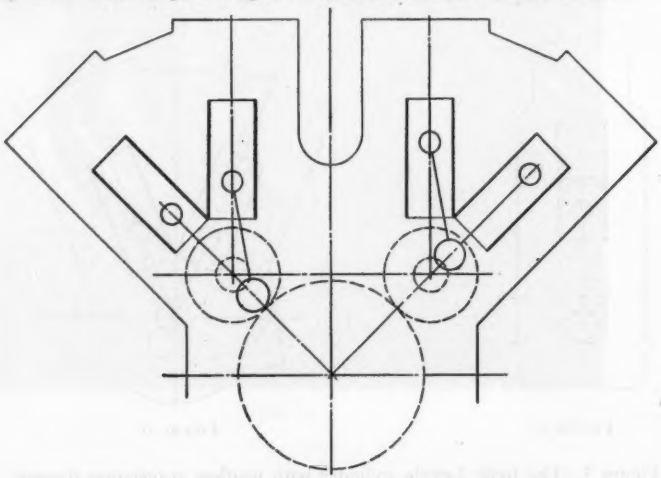


FIGURE 8.

Figure 8.—Double V arrangement with a pinion between each pair of 4-throw crankshafts in tandem engaging a bull gear on the driving shaft. Used in another 32-cylinder engine.

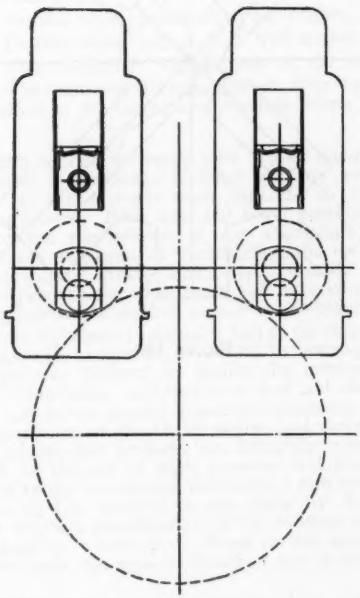


FIGURE 9.

Figure 9.—The quad or 4-engine unit of 24 cylinders has also a pinion between each pair of crankshafts in tandem, engaging a bull gear on the driving shaft.

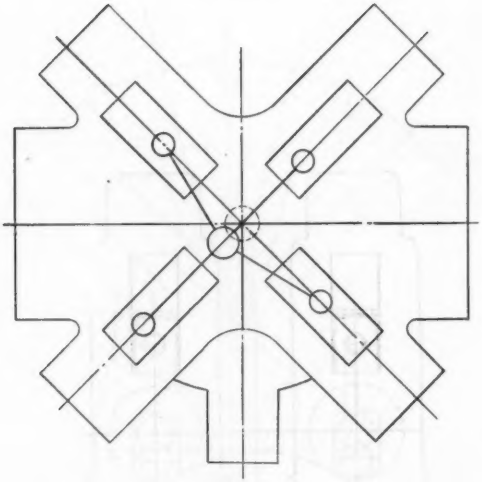


FIGURE 10.

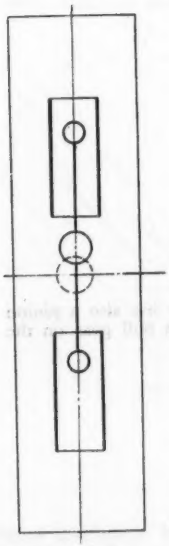


FIGURE 11.

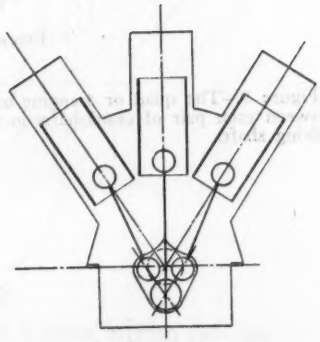


FIGURE 12.

Figure 10.—The pancake arrangement of 4 cylinders in the same plane on a single pin of a vertical crankshaft. The pistons of a 2-cycle engine being always under gas pressure, the connecting rods have been equipped with "slipper" ends and four of them are clustered around the same pin.

Figure 11.—Another vertical crankshaft engine with opposed cylinders producing a narrow unit of great accessibility.

Figure 12.—Three-bank or W arrangement with two link rods on a master connecting rod.

Figure 13.—Normal type of aircraft radial engine.

Figure 14.—Double-acting engine of a well-known type.

Figure 15.—Two-crankshaft arrangement of the opposed-piston engine.

Figure 16.—The diamond arrangement of four opposed-piston engines with eight pistons in the same plane through center of each crank.

Use of Gears became necessary with higher engine speeds and to the dismay of many a pessimistic prophet was not accompanied by either tooth failures or the excessive wear expected to result from the shock of cylinder explosions. Both spur and bevel gears have been successfully employed for the transmission of the crankshaft torque. With gears vindicated of any suspicion of unreliability, the use of lightweight high-speed engines will be greatly accelerated. Geared aircraft engines have been standard practice for some years and have contributed to the development of a lightweight engine as well as an efficient propeller.

Fuel Injection Equipment originally had to be classified as expendable or replaceable material to prevent tinkering by incompetent hands. Therefore its design was "frozen" to reduce the number of spares required. This lack of competition and incentive will and does retard its development. The cost of the present injection equipment represents too high a part of the total cost of the Diesel engine and therefore new and simpler methods of solving this problem are definitely needed. Maybe the unit injector with its absence of high pressure fuel lines could be simplified by elimination of the mechanical drive and a ram actuated by the cylinder pressure used in lieu thereof, as was done by Arschaouloff for larger engines. Or maybe a modification of the constant pressure system with a distributor deserves a new deal. Some of the lowest fuel consumptions were obtained with this type of injection and it still has many advocates in Britain.

Combustion Research was also eliminated to a great extent under the pressure of war with the result that combustion chamber characteristics which would not have survived normal competition continue in use. To provide an incentive for improvement, Figure 16 gives a curve of best fuel consumptions at rated power and speed obtained on shop tests of engines of widely varying speeds—700 to 2600 Rpm. It may be remarked here that the high fuel consumptions at higher speeds can hardly be explained by speed alone and apparently primarily reflect the lack of combustion research.

Ballistic effect of exhaust as an aid or even the principal means of inducing supercharging has not been thoroughly appreciated despite the efforts of Buechi, Kadenacy and others. Recent test results with a Buechi turbo-charged engine seem to have indicated that his insistence on uncooled exhaust pipes between exhaust valves and turbocharger to retain the temperature of exhaust gases is not the correct explanation of his excellent results since they were duplicated by a water-cooled exhaust header provided with gas-tight passages for each pair of non-interfering cylinders. Greater utilization of the kinetic energy in the slug of exhaust

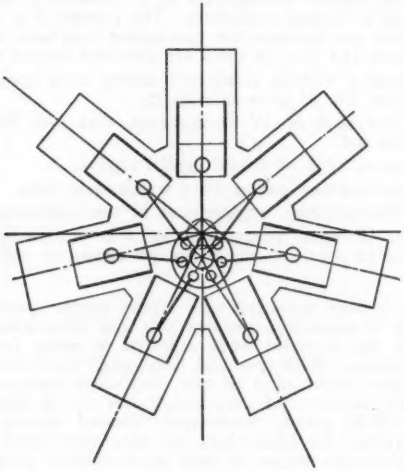


FIGURE 13.

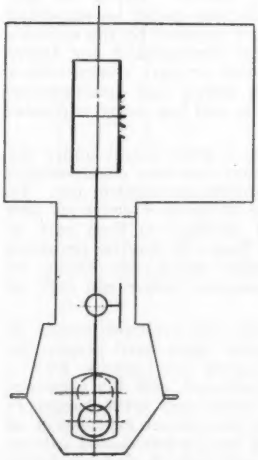


FIGURE 14.

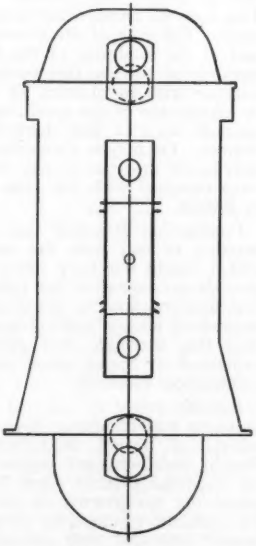


FIGURE 15.

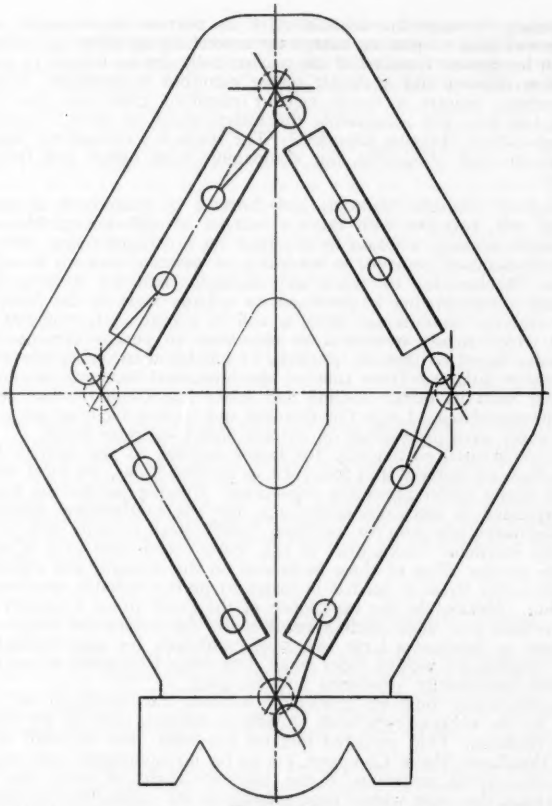


FIGURE 16.

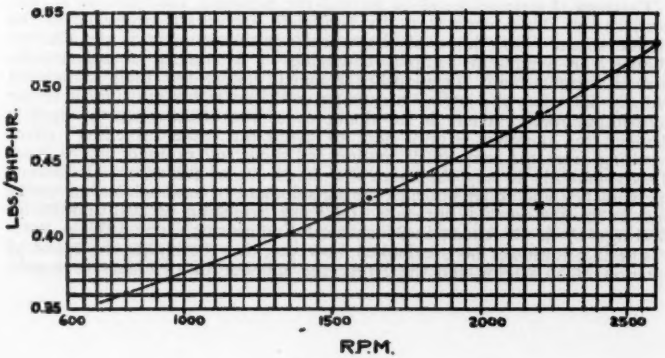


FIGURE 17.

gases passing through the exhaust valve or port at the moment of its opening will lead to greater charge or supercharging effect of cylinders and with less power required of the mechanically driven blower or even if there is no blower and a similar intake manifold is provided. Practical considerations require a single exhaust manifold even for 16-cylinder engines, but who can unscramble the interferences of these 16 cylinders with each other's charging processes? The result is a cut-and-try solution, far from the best obtainable, but comparable with others and therefore tolerable.

Scarcity of Strategic Materials and shortage of manpower, as another result of war, brought with them a number of different problems and have taught us many a lesson in economy on a national scale. Salvage, a heretofore despised concept, is becoming of greatest concern to all war activities. Reclamation of worn and damaged parts by welding, metal spray and electro-plating is practiced on a large scale in the Navy and further research on different items is still in progress. It was this same impetus which finally overcame all objections to porous chrome-plating of cylinder liners, so that the problem of suitable wearing surface can be permanently divorced from that of the structural strength required. A steel liner with its better cooling and easier lubrication is thus feasible when chrome-plated. Large Government and private facilities are now in existence for mass-production of chrome-plated cylinder liners.

Steel crankshafts, particularly for larger engines, of the desired degree of soundness of material and tolerance as to dimensions, are most difficult parts to obtain under prevailing conditions. Existing production facilities were expanded to meet this emergency, but the fundamental scarcity of low-phosphorus pig iron to produce "acid" steel remains and with it come the inevitable "inclusions" of the "basic" steel used as a substitute. Research on the effect of these inclusions on the strength and rigidity of the shaft under stress is needed in order to perfect suitable standards for inspection. Meanwhile, the experience of the Ford Motor Company, who claim to have cast eight million crank-shafts for automotive engines, has been used to produce a large cast steel crankshaft for experimental purposes. Crankshafts welded from small drop-forged component parts have also been successfully produced and installed.

A mechanically built-up crankshaft without the handicap of added weight of the existing type with shrunk-in journals may be the solution of this problem. One patented method has been used in small engines by the Waukesha Motor Company, but so far no equipment is in existence to cut the curved serrations of this joint for shafts of some size.

At present the most widely used remedy to the crankshaft bottleneck is the cast shaft from such patented compositions as Proferall and Meehanite.

Torsional Vibration seems to be still the perennial sub-topic of all discussions on Diesel engines and a legitimate excuse for profanity in the engine room. The "hope of the world" at the moment is the Chilton "bifilar" or pendulum damper, consisting of a number of segmentally shaped weights with ingeniously arranged rollers for their suspension radially-out when the engine is running and the weights are subject to the centrifugal force. There is no doubt as to the effectiveness of such a damper, the question is only how to avoid the "false brinelling" effect resulting from the very small motions of these rollers under heavy load. This apparently is very similar to the experience of the automotive industry with ball and roller bearings during shipment of automobiles on trucks and in freight cars. (Lubricants and False Brinelling of Ball and Roller Bearings—by J. O. Almen in Mechanical Engineering, June 1937, p. 415.)

The fact must not be overlooked, however, that the higher the speed of rotation of an engine and the larger the number of cylinders or crank-

throws it has, the more criticals will fall within the operating range and the more urgent therefore becomes the need of reliably controlling them.

Vibration and Noise are consequences of going to higher speeds. I believe we are approaching, if we have not already reached the point, where continuous presence of the operator in the engine room may be injurious to his health on account of noise. Since it is not at all necessary for the engineer to be able to lay his hand on the engine at all times and he can read instruments from a distance through a glass window just as well, there is no valid reason therefore why the modern steam practice should not be followed. The boilers are now installed in an airtight compartment under desired air pressure and separated from the adjoining room for auxiliaries and operating personnel under atmospheric pressure. Incidentally, the engine room could then serve as a charging air receiver and the engines installed therein would be automatically supercharged by the blower driven either by the engine itself or by a separate engine and discharging into the engine room. Automotive engineers will no doubt recognize in the above proposal an adaptation of the modern test cell used by the principal builders of high-speed engines. The idea, however, is too revolutionary for a seagoing engineer and he is still undecided as to his attitude toward it.

In conclusion I wish to emphasize that ship propulsion is very different from traction service, whether on rails or on highway, and even in the Navy, where full speed ahead is not continuous as in the merchant marine, two or three days at full speed are not an unusual war occurrence. Furthermore, cruising radius, which is proportional to the efficiency of fuel consumption, has a very real meaning in the Navy, quite independent and vastly more important than that suggested by the cost or even availability of fuel.

SHIPBOARD MAINTENANCE OF ROTARY PUMPS.

Mr. E. W. Chapman, Chief Engineer of the Blackmer Pump Company, is author of this eighteenth of a series of articles which have appeared in The Nautical Gazette, describing operation and maintenance of the various items of machinery common to marine installations. This article is reprinted from the November, 1943, edition of the Nautical Gazette.

Ships of all classes are being added to American fleets at an unbelievable rate. Among the auxiliary equipment on each of these vessels will be included power pumps of various designs, sizes and applications. For certain of these applications, marine designers have found rotary pumps can be used to better advantage than other types. This class of pumps will be found, to the practical exclusion of other designs, where self-priming and positive displacement pump characteristics are required, and where mounting space is limited. Care of centrifugal and reciprocating pumps has been well covered in an earlier paper in the "Keeping Shipshape" series. The present article will cover points that are of value to the operating engineer responsible for the care of shipboard rotary pumps.

Study the Installation.

Many pump troubles stem from faulty installation. It is therefore imperative that as soon as a new ship is put in service, those crew members in charge of pumps study and become thoroughly familiar with the installation. It must be kept in mind that the men who are to operate the ship have a much more personal and vital interest in this equipment than the men who installed it, or even the men who conducted the trials.

Many of the rotary pump applications aboard ship are connected to rather complicated piping layouts. A thorough knowledge of the valving, and the function of each valve is called for. It has happened too often to be laughed off, that considerable time is lost, and pump failure suspected, because one valve has been forgotten.

It frequently happens, in the case of rotary pumps for marine applications, that they are manifolded for dual applications. For example, a Diesel Fuel Oil Transfer Pump may be cut into the Diesel Fuel Oil Service piping, so that it may serve as a standby. Sometimes this results in possible emergency hook-ups, in which due to the friction losses involved, it would be impossible to get the rated capacity of the pump through the suction lines. This condition would result in noise and vibration in the pump, due to starved suction. Thorough familiarity with the pipe runs will enable the operating engineer to recognize such a condition, and perhaps do something about it. For instance, if the pump has a multi-speed drive, reducing the pump speed would help this condition, by reason of reduced delivery, and consequently reduced friction loss.

Piping and Pump Alignment.

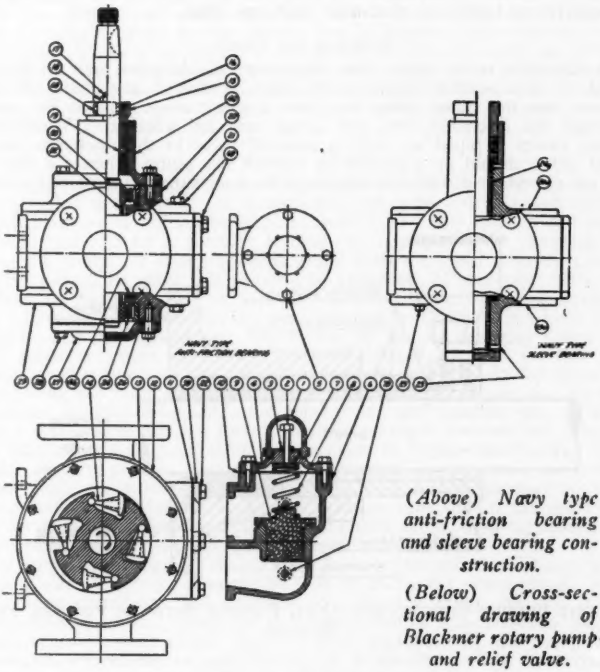
Study of the piping system will sometimes show that careless installation has produced dips or humps in pipe lines that will result in troublesome air traps. Such conditions should be changed wherever possible.

Probably the next most useful step will be to check pump alignment. Nearly every shipboard power pump is connected to its prime mover by means of a flexible coupling. If a separate gear reduction is interposed between pump and power, two flexible couplings will be used. Parallel alignment can be checked by using a straight edge across the rims of the coupling halves. Angular misalignment can be detected by inserting a feeler gauge between coupling halves, rotating the coupling 180 degrees, and trying the feeler gauge again. This measurement should not vary more than a very few thousandths of an inch. In most marine rotary pumps, especially those supplied to the Navy, internal bearings are used, with the packing outside the bearing, retained by a gland follower. In this arrangement, the follower fits the stuffing box too loosely to be of any value in checking alignment. One particular rotary pump design, however, features a sleeve bearing construction in which the packing is between the pumpage and the bearing. In this case, the gland is bushed to provide a bearing, and the fits are close, on the shaft and in the stuffing box. A considerable number of pumps of this design have found acceptance for marine applications. Where this design is encountered, pump alignment can be readily checked by removing the gland stud nuts, sliding the gland completely out of the stuffing box, and then sliding it back into the stuffing box. If the gland cannot be readily returned to its former position by hand, misalignment is present.

Correcting Misalignment.

If any misalignment is found, piping should be disconnected from the pump. If the mating flange on the piping springs away from the pump flange, or moves out of line with it, the piping should be adjusted to bring it into register with the pump connection. If when disconnected, the piping remains firmly in register with the pump flange, check to make sure that the piping is not too long, thus putting a thrust against the pump. When this condition is found, it is equally serious, and should be corrected. While the piping is disconnected, check the pump alignment again. If it now shows proper alignment, the piping alone is at fault; if misalignment is still present, it is probably due to distortion in mounting. This should be corrected by shimming. Every effort should be made to eliminate causes of misalignment, because it is a certain source

of trouble. Misalignment always results in heated bearings, and if neglected, bearing seizures. It also is reflected in accelerated pump and bearing wear, and in extreme cases, is the cause of actual breakage.



(Above) Navy type anti-friction bearing and sleeve bearing construction.

(Below) Cross-sectional drawing of Blackmer rotary pump and relief valve.

LIST OF PARTS

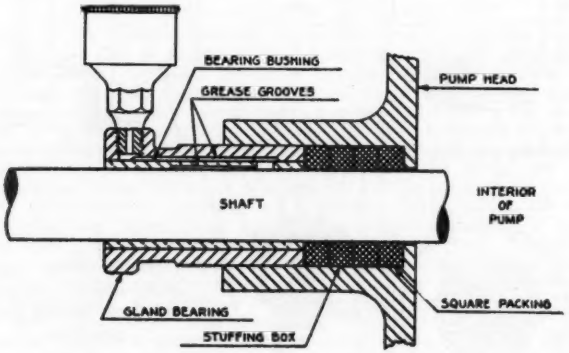
1 Bypass Cap
2 Adjusting Screw
3 Locknut
4 Bypass Cover
5 Cover Capscrews
6 Bypass Body
7 Spring Guide
8 Bypass Spring
9 Bypass Valve
10 Cover Gasket
11 Body Gasket
12 Cylinder
13 Rotor & Shaft
14 Bucket
15 Half Gland
16 Gland Bolts
17 Gland Studs
18 Gland Stud Nuts
19 Packing
20, 20a Inboard Head
21 Head Capscrews
22 Bypass Capscrews
23, 23a Outboard Head
24, 24a Bearing
25 Dowel Pin
26 Br'g. Cover Gasket
27 Bearing Cover
28 Covering Capscrews
29 Pipe Plug
39 Blanking Plate
40 Stuffing Box
46 Thrust Washer
65 Gland Nuts
69 Lockwashers
73 Pipe Plugs
82 Spacer Ring
83 Snap Ring

Direction of Rotation.

Before starting the pump, it is well to make sure that the pump has the desired rotation. Proper direction of rotation for a given pump is usually indicated by a cast arrow, or a decalcomania. Inlet and discharge ports are generally indicated as well, and it is well to verify that they are properly connected to known suction and discharge lines.

Starting the Pump.

In starting a rotary pump, one important consideration must be kept in mind. It is a positive displacement pump, so that if any liquid which is drawn into the pump does not have a free passage out of the pump through the discharge line, the pump will be wrecked. Whenever a rotary pump is piped up with a shut-off valve in the discharge line, a relief valve should be provided to protect the pump when the shut-off valve is closed. It is therefore important to make sure that if there is a shut-



Sleeve Bearing Construction With Packing Between Pumpage and Bearing.

off valve, there is also a relief valve. Relief valves may be built into the pump, attached to it, or piped into the discharge line entirely separate from the pump, and at greater or lesser distances from it. In the occasional installation where no relief valve is provided, but in which the discharge line can be closed off, always be sure to open the shut-off valve in the discharge line before starting the pump.

Leaks in Suction Line.

After the pump is started, note whether or not the delivery or rate of discharge is in accordance with expected performance. The most common cause of the failure of a new pump to deliver rated capacity is an air leak in the suction line. Suction lines must be absolutely air tight. A very small leak will allow air to be drawn into the system. A bubble of air under high vacuum will expand large enough so that the pump discharge will be materially reduced. This condition also keeps the pump filled with a mixture of air and liquid which is not homogeneous. The pumpage goes through the pump in varying densities, causing an erratic

pump action, accompanied by noise and vibration. This results in excessive pump wear, and in extreme cases, pump breakage. When this situation is encountered, the suction line should be emptied, disconnected from the pump, the end blocked, and compressed air introduced into the line. A soap solution or oil applied to all joints and around all valve stems should indicate by bubbling, the points of leakage.

If the pump is not delivering any liquid, or less than rated capacity, but a vacuum gauge in the suction line indicates a high vacuum, make sure that all valves between the pump and source of supply are open. If the valves are open, check any strainer that might be in the suction line for clogged strainer element or too fine a mesh. If this is not the cause, there is probably some foreign matter clogging or restricting the line. This must be located and removed.

Relief Valves.

It might happen that the operating engineer is satisfied that the pump is being filled with liquid, the suction side has been checked, and still the pump delivery is not up to rated capacity. The best clue to the answer to this one will be a pressure reading, on a gauge in the discharge line near the pump. If this reading is lower than the rated delivery pressure of the pump, the relief valve should be examined. Most poppet type relief valves are regulated by an adjusting screw, the setting being protected by a lock nut. It sometimes happens that this lock nut has not been set up tightly, and has backed off. Tightening up on the adjusting screw until the pressure gauge shows the pressure reading stamped on the relief valve, should result in the desired increase in delivery. In the event that tightening the relief valve adjusting screw does not increase the discharge pressure reading, disassemble the relief valve and examine the valve and valve seat. Some foreign matter may have lodged between the valve and its seat, allowing a portion of the pumpage to bypass continually. If the valve or valve seat has been damaged or scarred in some way, preventing a good seal, the valve may be ground in in the same manner in which an automobile engine valve is seated. At no time should the relief valve be set at a higher pressure than that stamped on it, without making sure the prime mover will not be overloaded.

If all of the foregoing checks have been made, and the delivery is still below rated capacity, it would be well to check the pump speed. Under the same conditions, the delivery of a Rotary Pump will be directly proportional to the speed. It is unlikely that a self-contained pumping unit, complete with power, that has been assembled by the pump manufacturer, will offend in this respect. It is always possible, however, that an error has been made. This is still more a possibility if only the pump has been purchased and later assembled with separately purchased prime mover on shipboard.

Sleeve-Bearing Pumps.

On new pumps, sleeve bearings should be watched closely until the operator is satisfied that they have been conditioned or "run in". In this respect pumps are no different than new automobiles or other mechanical equipment.

If a new bearing is running hot, something should be done about it before damage occurs. It is assumed that the alignment has been checked. Check to see that the packing has not been drawn up too tightly. If this has been corrected, and the bearing does not cool down, it may be necessary to have an operator stand by and grease the bearing continually until it can take the load successfully. This means that if the bearing is served by a grease cup, it should be given a half turn every thirty seconds. Sometimes a mixture of semi-fluid grease, with one part by weight of

powdered mica will help to condition a bearing. This is especially true where for some reason, the bearing or journal has been scored or roughened.

Anti-friction bearings do not usually present as much of a problem in rotary pumps as in centrifugals, due to the substantially lower speeds at which rotary pumps are run.

Packing.

Packing can be mentioned only in a general way. There are so many types and grades, each adapted to specific applications, that only when the service is known, can a recommendation be made. Generally speaking, the least amount of packing should be used that will sufficiently seal the gland against leakage. The operator should avoid "reefing down" or taking up too tightly on packing; it will burn the packing, and score the shaft, and will leak much sooner than packing which is more carefully handled. When the pumpage is of such a nature that slight leakage does no harm, packing should be taken up so that a drop or two a minute will seep through. "V" type packings are extremely sensitive, and should be tightened only in accordance with instructions coming with the packing. When packing is being replaced, joints in adjacent rings should be staggered with respect to each other. It is good practice to replace packing as soon as it is apparent that the customary amount of tightening does not stop leakage. It is also good practice to replace all the packing at this time, rather than to add a ring or two to the old packing.

DEVELOPMENTS IN PROPELLER DESIGN AND MANUFACTURE FOR MERCHANT SHIPS.

This paper was read by L. C. Burrill, Naval Architect of the Manganese Bronze and Brass Company, before the Institute of Marine Engineers on 24 August, 1943. It is reprinted from the Transactions of the Institute for October, 1943. The author has been responsible for the design of the propellers for many important and successful vessels, and his remarks elicited considerable discussion at the meeting.

"Aerofoil" Type Sections.

The most noticeable change in the design of propellers for merchant ships has been the general adoption of streamline blade sections of special type, based on the results of wind tunnel research in the related field of aerodynamics.

These new blade sections are commonly referred to as "aerofoil" sections to distinguish them from the "segmental" or "round-back" sections almost universally employed in earlier designs, although this earlier type of section is in fact a special type of aerofoil or "lifting plane" which has certain definite advantages in relation to pressure distribution, and in ease of definition for purposes of manufacture.

"Aerofoil" sections are characterized by the fact that the maximum thickness ordinate is forward of the middle of length and that the nose or leading edge is almost always thicker and more rounded than the tail or trailing edge. These features have been adopted because wind tunnel experiments show a favorable increase in the ratio of lift to drag at moderate angles of incidence, such as are applicable to the blade sections of marine propellers working under normal slip conditions.

Figure 1 shows lift and drag curves for typical "aerofoil" and segmental sections of several thickness ratios covering the range of usual

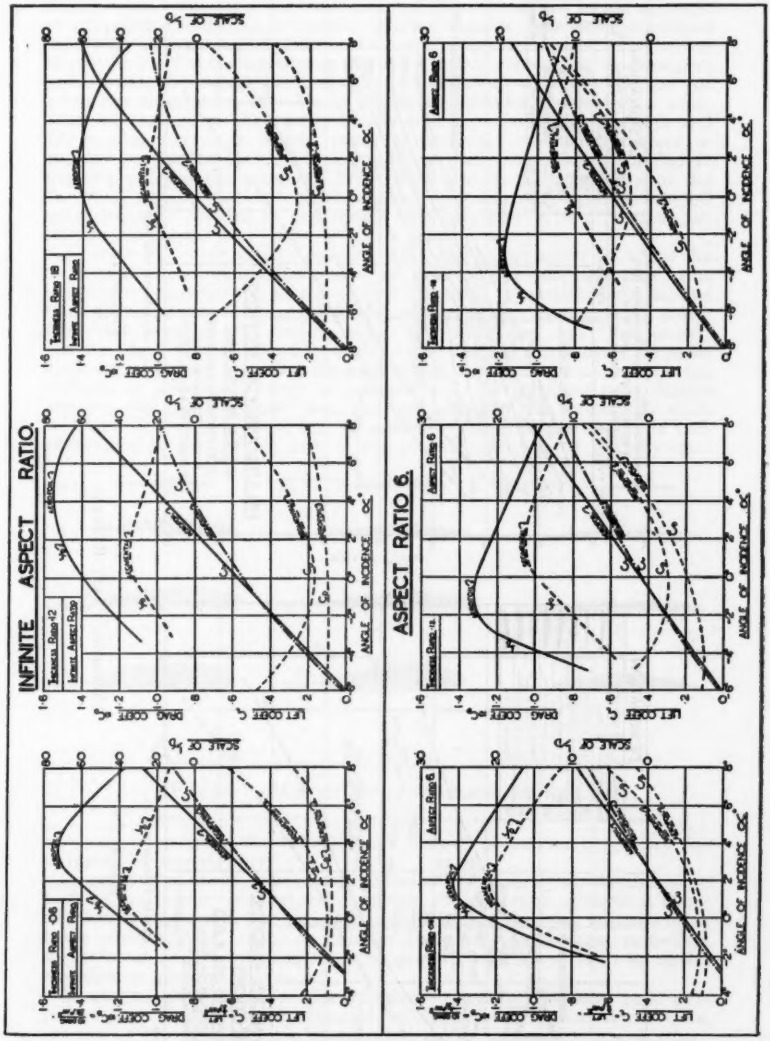


FIGURE 1.—LIFT/DRAG CURVES FOR AEROFOIL AND SEGMENTAL SECTIONS.

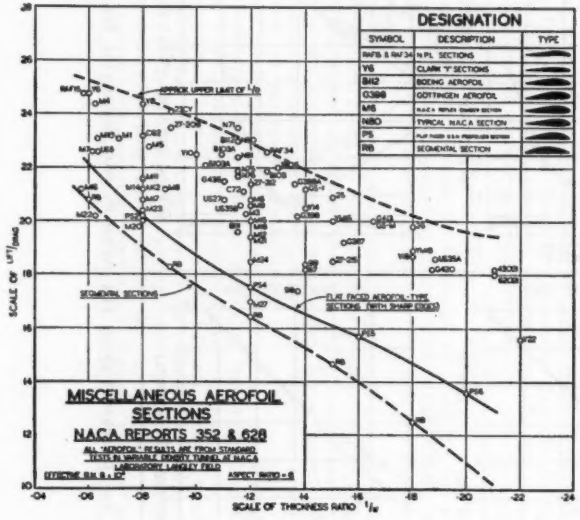


FIGURE 2A.

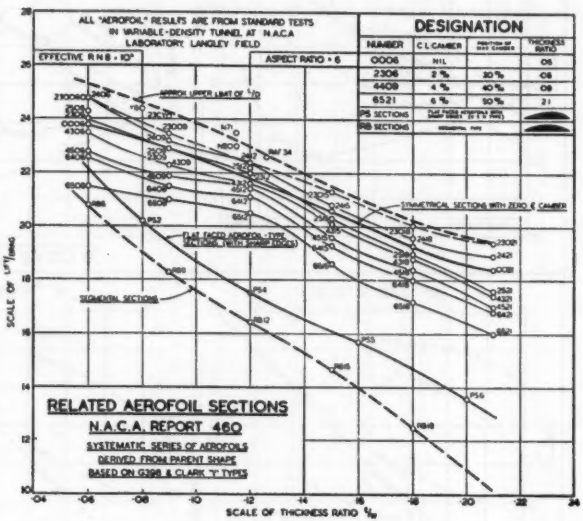

ALL "AEROFOIL" RESULTS ARE FROM STANDARD TESTS IN VARIABLE-DENSITY TUNNEL AT N.A.C.A. LABORATORY, LANGLEY FIELD
EFFECTIVE R N B = 10⁶
ASPECT RATIO = 6
DESIGNATION
NUMBER
C.L. CAMBER
POSITION OF MAX. CAMBER
THICKNESS RATIO
FS SECTIONS
RB SECTIONS
SYMMETRICAL SECTIONS WITH ZERO C.L. CAMBER
FLAT FACED AEROFOIL-TYPE SECTIONS (WITH SHARP EDGES)
SEGMENTAL SECTIONS
RELATED AEROFOIL SECTIONS
N.A.C.A. REPORT 460
SYSTEMATIC SERIES OF AEROFOILS DERIVED FROM PARENT SHAPE BASED ON G398 & CLARK "Y" TYPES
SCALE OF $L^{1/3}/D^{1/3}$
SCALE OF THICKNESS RATIO $^{t}/_{l}$

FIGURE 2B.

"AEROFOIL" TEST RESULTS.

thicknesses. In the upper diagrams the results have been corrected for infinite aspect ratio and therefore correspond to the values which are used in propeller calculations when the inflow velocities have been calculated separately by means of the vortex theory of propeller action. The curves shown in the lower diagrams on the other hand, correspond to an aspect ratio of six, this being the standard ratio of length to width adopted for wind tunnel experiments. These results are therefore useful for a practical comparison of the merits of different types of section shape, and lift to drag curves at various angles of incidence (measured relative to the flat portion of the underside of the sections), have been included in the diagrams for this purpose. From these curves, the superiority of the "aerofoil" type of section is quite evident and it can readily be understood why designers decided to experiment with this new type of section for marine propellers.

Since it was the common practice with marine propellers to adopt the same type of segmental sections at all radii of the propeller, this same procedure was followed in the earliest "aerofoil" propellers, and, in line with contemporary aeroplane practice, sections having the maximum thickness very close to the leading edge, and with very full "bull nosed" leading edges, were adopted throughout. In fact, in many cases the same basic aerofoil shape was used for each radius, the thickness ordinates being reduced or increased directly in proportion to the thickness ratio, despite the fact that the "aerofoils" used had been designed for moderately thick sections. It was also usual for all sections to have a flat underside corresponding to the driving face of the propeller, this being specially con-

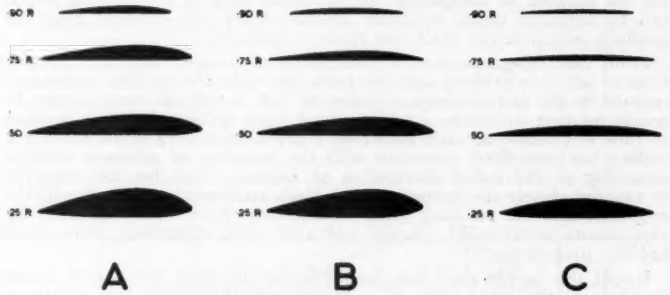


FIGURE 3.—MODERN PROPELLER-BLADE SECTIONS.

venient for defining the helical pitch surface of marine propellers for purposes of manufacture.

A number of quite satisfactory propellers were designed in this way, both in this country and on the continent, the new type of sections being substituted in place of the earlier round-back type, without adjustment in blade surface, pitch ratio, or other characteristics, and these propellers were found to give quite appreciable gains in efficiency, both in model experiments and in service.

It was however found in practice that in some cases these new propellers gave rise to undue vibration, while in others quite severe local erosion or pitting of the blades occurred under loading conditions where cavitation would not normally have been expected, and in some instances the revolutions in service did not agree with the intended design conditions.

From this early experience it became evident that while the "aerofoil" type of section showed definite gains in efficiency, care must be exercised in the choice of sections adopted for marine propellers working under different loading conditions, and also for the sections at different radii of the same propeller. It is therefore now quite usual for the section shape to vary considerably from root to tip of the same propeller, and for quite different types of sections to be adopted in different propellers, according to the loading conditions. (See Figure 3.)

In this connection, Figure 2 shows the maximum lift to drag ratios obtainable with a large number of "aerofoil" types, plotted to a base of thickness ratio. The left-hand diagram includes results for a large number of miscellaneous "aerofoils", and the right-hand diagram shows the values obtained for a systematic series of aerofoils, all derived from the same parent form, but having different degrees of centerline camber and with the maximum cambers at varying positions in the length. All the "aerofoil" tests shown were carried out in the same wind tunnel and under similar conditions, and the values have been taken from the test results for standard aspect ratio 6, after correction for tunnel interference. The values shown for segmental sections were not included in the original tests but have been obtained from other sources. These have been carefully checked from various tests and are believed by the writer to be consistent with the other results.

As mentioned in connection with Figure 1, the values shown in these diagrams must be corrected for infinite aspect ratio conditions when used in propeller calculations, but in the present form they are very suitable for the purpose of comparing the maximum ratio of lift to drag which can be expected under optimum conditions for the different types of aerofoils and different thickness ratios considered.

From these diagrams, one or two points of interest may be observed. First of all, it is evident that the most favorable lift to drag ratios correspond to the lower thickness ratios, so that in general terms it may be concluded that advantage is to be gained from making the blade sections as thin as possible at each radius of a given propeller, when the blade surface has been fixed, consistent with the provision of adequate strength according to the radial distribution of loading. This has led designers to examine closely the strength of the blade sections at each radius of the propeller, and the resulting gains in efficiency have been confirmed by experiments with model screws, and also from experience with actual full-size propellers.

It will also be observed that for the lower thickness ratios, such as are common in the outer parts of a marine propeller blade, the difference between the lift to drag ratios obtainable with various section shapes is much less than for the larger thickness ratios at the other end of the diagram, such as would correspond with the root portions of the blades. Little advantage is therefore to be expected from the adoption of special aerofoil shapes near the tip of the blades, and it has been found advisable to adopt sections resembling more closely the usual round-back type in this region of the propeller. It may also be noted in passing that this is specially true in the case of thin wide-bladed propellers, such as are usually fitted to naval vessels.

For the thicker sections of about ·20 thickness ratio, sections having a relatively low centerline camber and a rounded face are more favorable than those having a flat face and high centerline camber, and experience has shown that this is correct. For example, this is confirmed by the series of propellers tested by Dr. G. S. Baker and discussed in his papers published in the 1932 and 1934 Transactions of the Institution of Naval Architects, and by other model tests. In the papers mentioned above,

the feature referred to is described as wash-back, and in propeller practice is also known as turn-up of the driving face, and it has been found that sections having turn-up corresponding to a centerline camber of between ·020 and ·030 are most satisfactory for the root sections of mercantile propellers of normal proportions.

Recent work has in fact emphasized the importance of considering separately the basic centerline camber of the section, and the streamline shape with which this centerline may be regarded as being "clothed", rather than to pay attention to mere geometrical variations in shape. In very simple terms, the streamline contour may be compared to the longitudinal section of a fish, having in most cases the characteristic fuller forward body and fine tail, while the cambered centerline may be taken to represent curvature of the backbone. It is evident that different characteristics will be obtained by altering either the streamline contour or the cambered centerline in any particular case. For example the systematic series of aerofoils mentioned in connection with Figure 2 were obtained in this way from a single parent shape, by varying the amount and position of maximum centerline camber.

Analyzing the characteristics of a large number of sections in this way, it has been found that very satisfactory marine propeller sections may be designed by employing suitably curved centerlines having a camber of about ·020 to ·030, as mentioned above, with the maximum camber approximately 40 per cent of the length from the forward end, and adopting streamline contours with the maximum thickness at about 33 per cent from the leading edge.

For purposes of design and comparison, and to ensure fairness of both centerline and streamline shape, it has been found convenient to draw these out in a standard form, similar to that adopted for recording sectional area and waterline curves for ship forms, on a percentage basis. As an example of the method adopted for this purpose, the centerline cambers and streamline contours corresponding to the typical propeller sections shown in Figure 3(B) have been reproduced in Figure 4. From this diagram, it will be seen that both the cambered centerlines and the streamline shape are varied in a regular manner from root to tip.

The optimum centerline camber for any section depends very largely on the lift coefficient at which the section will be working in the average design condition. For example, the appropriate centerline camber for

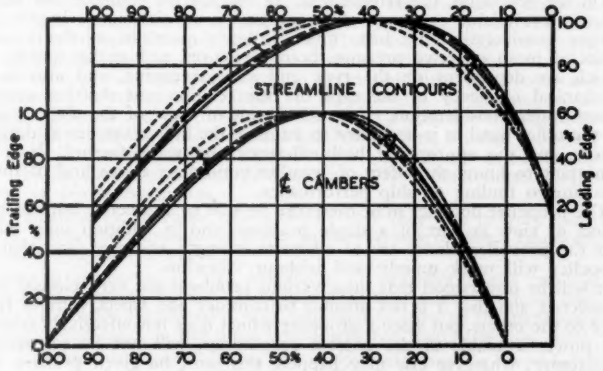


FIGURE 4.—SECTION ANALYSIS DIAGRAM.

typical root sections at about ·25 radius may be taken as equal to ·13 times the working lift coefficient obtained after due correction has been made for blade interference or "cascade" effect, and it will be found that this corresponds very closely with good modern practice. For the tipmost sections of marine propellers, the centerline camber which can be provided is frequently less than the optimum value, unless the blade sections are given a hollow face, and since this is not favored owing to difficulties in manufacture, such sections are usually designed with a flat driving face.

So far as the streamline contour is concerned, this has a considerable influence on the local velocities round the section, and consequently on the distribution of pressure and suction on the face and back of the blades. In general terms, it may be said that increase in fullness of the forward part of the section, or movement of the maximum thickness towards the leading edge, will increase the suctions on the forward part of the back of the section and decrease those on the after part. For a given lift coefficient the value of the maximum suction will also be increased and the drag coefficient will be slightly decreased.

This increase in suction on the forward part of the back of the blades is in fact a well-known feature of the new "aerofoil" type sections, and it can be shown by calculation that this contributes to the increased efficiency obtainable with this type of section, since the resultant lifting force is moved towards the direction of motion or thrust. The task of the designer is therefore to control the distribution of velocity round the section, and to decide how far the suction on the forward part of the blades can be increased in any given circumstances without risk of instability of flow, and without exceeding the safe limit of maximum suction from the point of view of cavitation. This will be referred to later in the section dealing with this subject.

Design and Analysis Methods.

The propeller is the link between the engine room and the ship, and both the marine engineer and the naval architect are interested in its satisfactory performance as a propelling unit. The chief concern of the former is, of course, that the propeller will be able to absorb the designed horsepower at the correct revolutions, while the latter is mainly interested in the ship speed which can be obtained for given engine size and power.

On the one hand, the requirements of the marine engineer can be met when the available torque at different engine revolutions is known, and proper consideration has been given to such questions as the practical limits of mean effective pressure, boost pressures, and piston speeds, etc., which are dependent on the type and size of engine, and also to the mechanical efficiency of the type of main engine and shafting arrangements under consideration. From the point of view of the ship designer on the other hand, it is necessary to estimate the hull resistance at different speeds, and the appropriate hull efficiency and wake factors. It is also important to know the effect of weather conditions at sea and of roughness due to fouling on ship performance.

The propeller designer must therefore be able to appreciate both of these points of view as part of a single problem, and in addition must ensure that the propeller blades are of adequate strength and area, and that the propeller will work quietly and without vibration.

It will be understood that these various problems are very closely interconnected, and that it is not possible to consider one aspect without reference to the others, but since a propeller which does not effectively transmit the power available at the correct revolutions, will not be regarded as satisfactory, whatever else may happen, this must be given primary consideration, and must be kept in mind throughout the various calculations.

The most practical method of approach from the propeller designer's point of view is therefore to decide first of all the amount of power which will be available and the corresponding revolutions at which this can be developed, and then subsequently to estimate the ship speeds which this engine horsepower will give under various conditions of loading and weather. The final propeller dimensions chosen should be such that the highest overall efficiency will be obtained in several typical conditions of loading, under average weather conditions at sea.

Since this will entail the examination of a number of alternative conditions for any given case, it is important to have at hand the necessary means for determining quickly the power which will be absorbed and the propeller efficiency obtainable under different conditions of slip and propeller loading.

The most satisfactory method of obtaining the data required for this purpose, is to analyze actual sea performance data for a large number of ships of different types by means of standard design and analysis charts derived from systematic experiments with related model screw propellers in open water, and to express the results of such analysis in terms of the usual coefficients and factors employed in connection with model experiment tank work; that is to say, it is convenient to obtain in each case an "analysis wake fraction" and an "analysis hull efficiency" which may be compared directly with the corresponding figures from tank experiments.

In this way, mean values of the various factors can be obtained and tabulated in groups corresponding to different ship types and sizes, and all the basic design data so collected can be related back to a single diagram, or set of diagrams, which can then be used with confidence for new design work, or for interpreting the results of ship model tests when these are available.

During the last fifteen years a considerable amount of valuable information regarding the performance of propellers of different types has been published by various research stations, and standard series results are now available for both round-back elliptical propellers and "aerofoil" propellers of modern merchant-ship types, covering a wide range of slip conditions.

In this connection, special mention should be made of the work of the late Admiral D. W. Taylor, published mainly in the Transactions of the American Society of Naval Architects, including the composite diagrams for round-back propellers based on propeller experiments, by Froude, Schaffran, Taylor and Durand, and the A, B, C and D series for special section types, and also of the valuable series of diagrams for 3 and 4 bladed propellers of recent merchant-ship types published by Troost and Van Lammeren of the Dutch Tank at Wageningen, which are to be found in the Transactions of the North East Coast Institution of Engineers and Shipbuilders. In the field of wide-bladed propellers, suitable for naval vessels, there is also the extended Froude series carried out at Haslar by Mr. R. W. L. Gawn and published in the 1937 Transactions of the Institution of Naval Architects.

An examination of all these results shows that they are in very close agreement, when due allowance is made for the various thickness fractions, blade shapes and section types.

For ease in manipulation and simplicity in calculation, the method introduced by Admiral Taylor of representing the results of these tests in terms of the basic variables Bp. and δ, plotted relative to the parameter "a", or face pitch-ratio, is the most convenient for merchant ship work.

Figure 5 shows a composite design and analysis diagram for four-bladed propellers obtained by plotting curves of constant Bp. to a base

of $\delta = \frac{Nd}{Va}$ with a vertical scale of face pitch-ratio, and adding a series of curves for constant K_q the variable most frequently used in aeronautical work. To obtain more suitable numerical values for marine propeller work, these latter curves have actually been numbered in terms of an analysis coefficient A_q which is equal to $1000K_q$ and is given by the following expression:—

$$A_q = 1000K_q = d^2 \frac{9{\cdot}46P}{\left(\frac{Nd}{1000}\right)^3}$$

where P=delivered horsepower, d=diameter in feet, and N=revolutions per minute. The values of Bp. shown in the diagram are those published by Admiral Taylor at the World Engineering Congress held in Tokyo in 1929, which have been found to be very satisfactory in practice over a number of years.

Design Calculations.

In using this diagram for design purposes, it is usual to enter the diagram with the appropriate value of B_{p4} and to choose the pitch ratio corresponding to optimum efficiency, which is given by the line 00. This basic variable B_{p4} is given by the expression:—

$$B_p{}^4 = \frac{NP^{\frac{1}{2}}}{Va^{2\frac{1}{2}}}$$

where Va=speed of advance in knots, P=DHP and N=revolutions per minute, as above.

It will be seen that this includes all the basic design conditions mentioned at the beginning of this section (i.e., speed, power and revolutions), so that for the point of intersection of the appropriate B_{p4} line with the heavy line 00, the main characteristics of the best propeller can be read off from the scales of pitch ratio and "δ". The diameter is then given by $d = \frac{\delta . Va}{N}$, and the face pitch by $p = a \times d$.

If the diameter obtained in this way is too large for reasons of clearance or immersion, then the most suitable dimensions can be determined by calculating the "δ" value corresponding to the maximum diameter which can be fitted, and reading off the pitch ratio corresponding to the already determined value of B_{p4}. In all cases the propeller chosen should plot to the right of the line 00.

For any position in the diagram, the corresponding propeller efficiency can be obtained from the efficiency curves shown by dotted lines, and if the efficiency thus obtained differs from that required to obtain the speed of ship assumed, it will be necessary to adjust the speed accordingly, and repeat the above operations until proper agreement is reached.

Analysis-Wake Calculations.

For analysis purposes, the value of the ship speed, power and revolutions, together with the face pitch-ratio of the propeller, are known. It is therefore a very simple matter to calculate the value of A_q and enter the diagram along the appropriate pitch-ratio line, the corresponding "δ" value being read from the horizontal scale.

The analysis "Va" is then given by the simple expression $Va = \frac{N.d}{\delta}$ and the analysis wake fraction is obtained from the formula

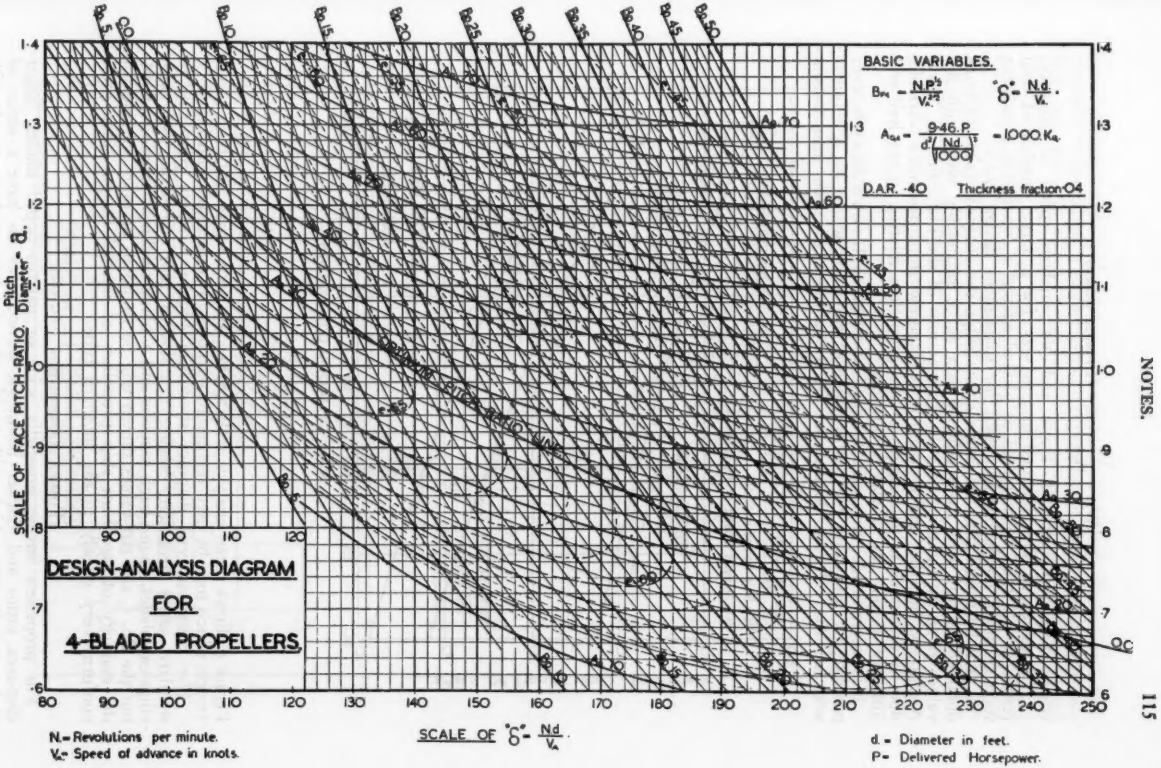


Figure 5.

NOTES.

$$W_F = \frac{Vs}{Va} - 1 \quad \text{(Froude wake)}$$

where Vs=ship speed in knots and Va=analysis speed of advance.

The values of W_F obtained in this way for a large number of ship results should then be tabulated or plotted against a convenient parameter such as block coefficient and used for subsequent design work.

It will be found that for single screw ships the analysis wake fractions obtained in this way from reliable data agree reasonably well with the values given by Luke in the 1910 Transactions of the Institution of Naval Architects, but that the results for twin screw ships are more variable, due to variations in positioning of the propellers relative to the hull and different shapes and angles of bossing.

It is recommended that each designer should obtain his own data wherever possible, by analyzing the performance of ships for which he is familiar with all the relevant details, but in the absence of such information,

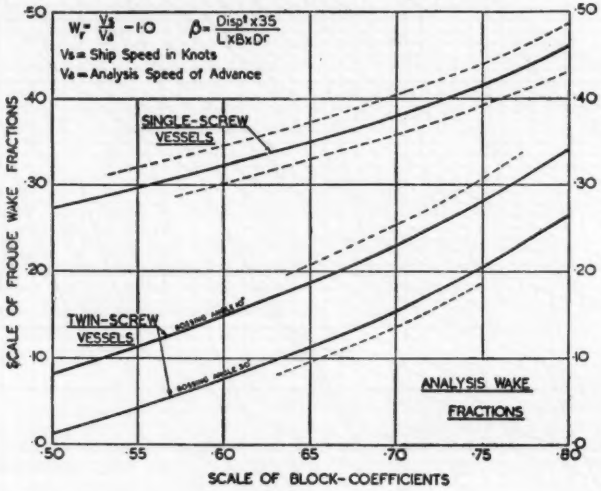


FIGURE 6.

Figure 6 shows mean analysis wake curves for both single and twin screw vessels which may be used for design purposes in connection with Figure 5.

The design-analysis diagram shown in Figure 5 refers primarily to standard series propellers having a disc-area ratio of about ·40, with elliptical blades, round-back sections and a blade thickness fraction of ·04, but the Bp, δ, and Aq values shown will be found to be quite suitable for the majority of merchant ship propellers of normal design, since these conform very closely to the standard type.

POWER CORRECTION FACTORS.

For propellers having unusually wide or narrow-tipped blades, larger disc-area ratios and thicker blades, it is necessary to have a series of correction factors which will enable the diagram to be entered with equivalent dimensions. A convenient method of establishing such correction factors is to obtain first of all a power factor expressing the ratio between the delivered horsepower "P" developed by a standard screw of given main dimensions and the corresponding power "P_1" developed

by a similar screw with increased surface, wide-tipped blades, etc., and secondly to introduce a pitch-ratio correction factor based on the relationship between the effective or "analysis" pitches of different types of propellers.

The power correction factors for thickness-ratio, disc-area ratio and boss-diameter ratios are approximately linear for the range of variations covered by normal merchant-ship propellers, and may be expressed by the following empirical formulae obtained as a result of analyzing the available data on this subject.

(1) *Thickness-ratio correction*=1+5·5 (t/D−·040).

(2) *Disc-area ratio correction*=1+·53 (D.A.R.−·40).

(3) *Boss-diameter correction*=1−·50 (d/D−·20).

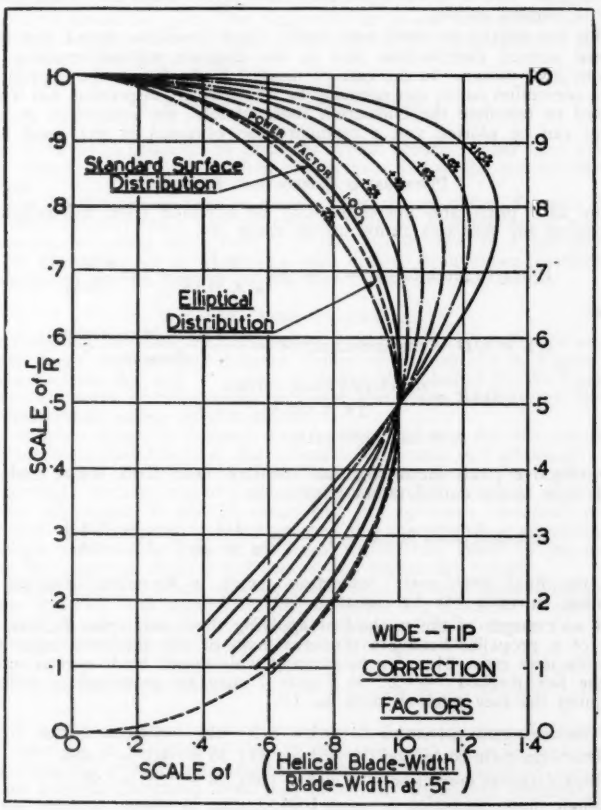


FIGURE 7.

In dealing with wide and narrow-tipped blades, it has been found convenient to draw out the radial distribution of blade surface in a standard diagram such as is shown in Figure 7. In this diagram, the helical blade width at each radius is expressed as a proportion of the width at ·5 radius, and the scales are so chosen that any elliptical distribution of surface is represented by a semi-circle with center at ·5R, independent of the disc-area ratio.

The shape marked 1·0 corresponds to the standard Taylor elliptical developed plan-form, and it will be found that most merchant screws lie between this line and the contour marked +8 per cent which represents approximately the Froude wide-tipped plan form for moderate disc-area ratios. To obtain the appropriate power correction factor for any given design it is necessary to plot the appropriate radial distribution of surface in the diagram and to estimate the correction factor from the values shown on the various curves.

This can usually be done very easily, since it will be found that most normal surface distributions plot in the diagram without crossing the various lines shown. In the case of unusual blade shapes, the appropriate mean correction factor can normally be estimated by inspection, but if it is desired to calculate the correction more closely, the correction at each radius can be plotted and a moment-mean obtained in the usual way.

Pitch-ratio Correction Factor.

The final pitch-ratio correction can be obtained from the following expression for disc-area ratios up to about ·75:

$$\text{Effective pitch factor} = 1 + \cdot 88\frac{\text{t. f.}}{\text{a} \times \text{DAR}} + \cdot 18\frac{\text{a} \times \text{t. f.}}{\text{DAR}} \qquad (4)$$

where

$$\text{t. f.} = \text{t/D} = \text{thickness fraction} = \frac{\text{thickness at centerline}}{\text{diameter.}}$$

$$\text{DAR} = \frac{\text{developed blade surface}}{\frac{1}{4}\pi \times \text{dia.}^2}$$

$$\text{a} = \text{face pitch-ratio.}$$

The effective pitch factor for the standard Disc Area Ratio, and the pitch ratio under consideration is given by:

$$\text{Standard pitch factor} = 1 + \cdot 88\frac{\text{t. f.}}{\text{a} \times \cdot 40} + \cdot 18\frac{\text{a} \times \text{t. f.}}{\cdot 40} \qquad (5)$$

and the final pitch ratio correction factor is therefore obtained by dividing equation (4) by equation (5).

As an example of the method of applying these correction factors, the case of a propeller having a disc-area ratio of ·55, thickness ratio ·046, boss-diameter ratio ·24 and a moderately wide-tipped blade corresponding to the line marked "+4%" in Figure 7 may be examined as follows, assuming the face pitch-ratio to be 1·0.

(1) Thickness-ratio correction $= 1 + 5 \cdot 5\,(\cdot 046 - \cdot 040) = 1 \cdot 033.$
(2) Disc-area ratio correction $= 1 + \cdot 53\,(\cdot 55 - \cdot 40) = 1 \cdot 080.$
(3) Boss-diameter correction $= 1 - \cdot 50\,(\cdot 24 - \cdot 20) = \cdot 98.$
(4) Blade shape correction $= 1 \cdot 04.$
$\therefore$ Power correction factor $= 1 \cdot 033 \times 1 \cdot 080 \times \cdot 98 \times 1 \cdot 04 = 1 \cdot 137.$

$$\text{Effective pitch-factor} = 1 + \cdot 88 \times \frac{\cdot 046}{1 \cdot 0 \times \cdot 55} + \cdot 18 \times \frac{1 \cdot 0 \times \cdot 046}{\cdot 55} = 1 \cdot 089.$$

$$\text{Standard pitch-factor} = 1 + \cdot 089 \times \frac{\cdot 55}{\cdot 40} = 1 \cdot 122.$$

$$\therefore \text{Pitch-ratio correction factor} = \frac{1 \cdot 089}{1 \cdot 122} = \cdot 97.$$

The equivalent face pitch-ratio for use in connection with Figure 5 is therefore $\cdot 97 \times 1 \cdot 0 = \cdot 97$ and the power obtained for any "δ" value is to be multiplied by 1·137. Assuming the propeller to be working in the region of optimum "δ"=140, the B_{p_4} value would be 12·42, and the B_{p_4} value for "a"=1·0 would be 13·0. The final ratio of power absorption for this propeller and a propeller of ·40 DAR, ·04 t/D, ·20 d/D and elliptical blade shape, is therefore given by:

$$\frac{P_1}{P \cdot} = \left\{\frac{12 \cdot 42}{13 \cdot 0}\right\}^2 \times 1 \cdot 137 = 1 \cdot 038.$$

That is to say, a propeller of this type would absorb 3·8 per cent more power than a propeller of standard proportions, for similar conditions of working.

It may be mentioned here in passing that the empirical formula for blade-thickness correction was determined from an examination of model test results, but since this effect is in fact mainly due to a change in effective pitch, formula 5, which has been derived from more theoretical considerations, can be used as an alternative means of calculation.

Aerofoil Type Propellers.

For designing "aerofoil" type propellers, it is usually convenient to use a separate design and analysis diagram similar to that shown in Figure 5, but based on the Bp. and "δ" values published by Admiral D. W. Taylor for the "D" series of Shipping Board models and the Wageningen B4·40 series mentioned earlier in this section.

The diagram shown in Figure 5 may, however, be used for this purpose, provided corrections are made for optimum pitch-ratio and efficiency.

Figure 8 shows the difference in positioning of the line 00 for segmental and "aerofoil" sections, and also the corresponding differences in optimum propeller efficiency. It will be noted that the optimum "aerofoil" propellers are slightly larger in diameter than the corresponding round-back propellers, and that the gain in efficiency varies from about 10 per cent at low Bp. values to about 5 per cent at the higher Bp. values.

So far as effective pitch is concerned, there is very little difference between "aerofoil" and round-back propellers, but if it is desired to estimate this effect in any given case, the following modified formula for the effective pitch of "aerofoil" type propellers may be used:

$$\text{Effective pitch factor} = 1 + \cdot 80 \frac{\text{t. f.}}{\text{a} \times \text{DAR}} + \cdot 16 \frac{\text{a} \times \text{t. f.}}{\text{DAR}} \qquad (6)$$

and the result compared with that obtained from equation (4) above.

For example, in the case of the ·55 disc-area ratio propeller examined in the foregoing section, the modified pitch factor for "aerofoil" sections becomes 1·080 as compared with the value 1·089 previously obtained, so that the effective pitch is reduced by about ·8 per cent.

For propellers having a varying pitch from root to tip of the blades, it is

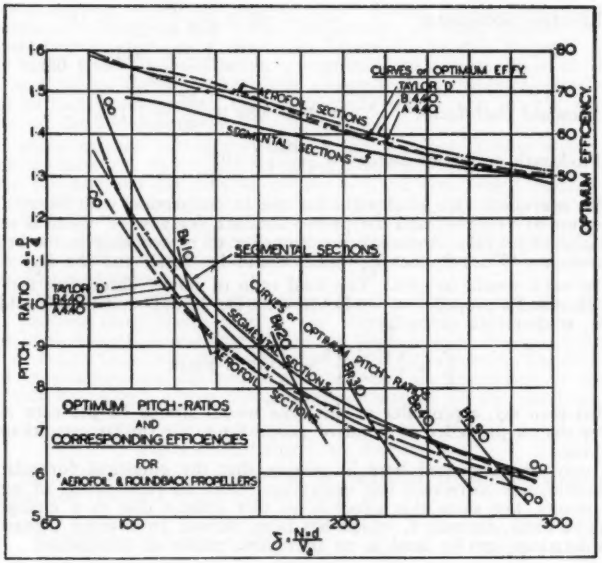


FIGURE 8.

usually sufficient to determine the equivalent mean face-pitch by applying a simple moment-mean method of calculation, but in special cases it may be desirable to calculate the no-lift angle and effective pitch for each section of the propeller, using more detailed methods of analysis.

GENERAL DESIGN CONSIDERATIONS.

Apart from the determination of the main dimensions (i.e., diameter, pitch and surface) and the choice of suitable blade sections, several other considerations must be given attention in designing a new propeller.

For example, it is important to secure the maximum clearances which can be obtained from rudders, fins, sternframe or shaft bossings by the adoption of suitable rake and skew-back of the blades, since in this way the natural flow of water through the propeller will be disturbed as little as possible, and the most favorable hull-efficiency will be obtained.

The risk of vibration will at the same time be reduced to a minimum, and in single-screw ships there will be less chance of interference with the natural formation of the tip vortices, thus reducing the possibility of local breakdown of the flow and/or local erosion in way of the blade-tips.

Advantage is also to be gained from the careful designing of the boss, and the fitting of a suitable streamline cone-cover and ropeguard, designed to obtain continuous streamline conditions of flow in way of the root-fillets.

Flow conditions in this region can also be improved in some instances by drawing out the root sections in proper relation to each other and modifying their shape in such a way as to avoid excessive venturi-effect between adjacent blades.

The best radial distribution of blade surface for the majority of merchant ship propellers approximates very closely to the standard Taylor blade shape, although it may be advantageous in some cases to extend the root-sections to occupy the full length of the boss and thus avoid excessive thickness-ratios, or to adjust the blade-widths in way of the tips to suit the conditions obtaining in this part of the blades, where the normal flow is modified by the formation of the tip-vortices.

With regard to pitch variation, it is now generally accepted that a reduction in pitch towards the root of blades is advantageous from many points of view. That is to say, more favorable blade widths and blade angles can be obtained in this way, and the radial distribution of loading is improved, especially in view of the normal variation in wake velocities.

In some cases, it may be considered desirable to reduce the pitch-angles in way of the tips, but for normal merchant-ship propellers this type of pitch variation is not favored by the writer.

In any given case, the pitch-variation finally adopted should be as simple as possible, and should also be continuous and "fair" from root to tip. This also applies to the shape of the blade sections, and it is therefore advisable to prepare a fairing diagram, somewhat on the lines of the usual ship's body-plan, to ensure that this condition is achieved in any new design.

Cavitation.

Propeller cavitation is a form of overloading in which the negative-pressures or "suctions" caused by the action of the blades exceed the positive-pressures provided by the static head of water over the propeller.

The result is that the water fails to follow up the blades completely, and cavities filled with air and water vapor are formed. These cavities may collapse on the blades and cause considerable "erosion" or damage to the metal surfaces, or may collapse behind the propeller, in which case there will be a marked loss in power absorbed, and also in efficiency.

Except in extreme cases, serious cavitation can usually be avoided by the provision of adequate blade area, and the adoption of blade sections which avoid the occurrence of high suction-peaks.

It has long been understood that the main factors governing the suitability of a design in relation to cavitation are thrust-loading and speed of rotation. Various cavitation criteria have therefore been suggested from time to time, expressed in terms of thrust per square inch of surface and limiting tip speeds.

In practice, this method leads to widely different values being adopted for various types of ships and propellers, and it is sometimes difficult to choose the appropriate limits for a new ship, especially in the case of new or intermediate types.

An attempt has therefore been made to develop a cavitation analysis method based on recent research work in this field, and embodying the necessary factors and criteria in such a form that the results for all types of ships can be plotted in suitable relationship to each other on a single chart or diagram.

When a single aerofoil is tested in a cavitation tunnel, the test results are usually expressed in terms of a cavitation number σ which is equal to the static head divided by the dynamic head $\frac{1}{2}\rho V^2$, and also in terms of the angle of incidence, or "lift coefficient", at which the section is working.

It was therefore decided to adopt as a base scale the value of σ for ·7 radius of the propeller, which is given by the expression:—

$$\sigma_0 = \frac{p - e}{qv + q_R} = \frac{p - e}{q_T} = \frac{p - e}{\frac{1}{2}\rho \cdot V_T^2}$$

where p=static head of water and atmospheric pressure,
e=vapor pressure,
ρ=density,
qv=dynamic pressure component in the direction of advance,
qR=dynamic pressure component in the direction of rotation,
qT=resultant dynamic pressure.

which is very similar to the base scale shown in the diagram included in the recent paper by Sir Stanley V. Goodall, published in the 1939 Transactions of the Institution of Naval Architects.

To obtain a suitable coefficient for use as ordinate in the diagram, which would take the form of a mean lift coefficient, it was decided to calculate the resultant of the thrust force and the torque force on the blades and divide this by $\frac{1}{2}\rho AV^2$ so that the coefficient became:—

$$C = \frac{R}{\frac{1}{2}\rho . A_d . V_T^2} \text{ where } R = \text{resultant force} = \sqrt{T_f^2 + Q_f^2}$$

ρ = density
Ad = developed surface
V_T = speed at ·7R

the appropriate expressions for Tf and Qf being respectively:—

$$Tf = \frac{THP \times 33{,}000}{101 \cdot 3 \times Va} \text{ and } Qf = \frac{DHP \times 33{,}000}{2\pi \times N \times \cdot 7R}$$

It was subsequently found from calculations for a representative number of propellers of different types that the value of R/Ad was approximately equal to T/Ap, where Ap is the projected surface, the values of R/Ad usually varying from about 1·05 to 1·07 times T/Ap.

In view of this, it was decided to adopt the expression T/Ap in place of R/Ad in order to simplify the incidental calculations the vertical scale finally used being given by:—

$$\tau_0 = \frac{T}{\frac{1}{2}\rho . Ap . V_T^2} = \frac{T/Ap}{q_T}$$

The diagram obtained in this way is shown in Figure 9 and it will be seen that the general trend of the values plotted for a number of propellers of different types is similar to that of the limiting lift-coefficient line given by Lerbs in his article published in "Werft Reederei Hafen", July, 1931, although the numerical values are of course not strictly comparable. In view of the fact that the more important heavily-loaded propellers are to be found at the lower end of the scale, a double logarithmic scale has been adopted, to increase the spacing in this region of the diagram.

From the data examined, three main lines have been drawn in the diagram, representing respectively the upper limits of τ_c which can be adopted for heavily-loaded propellers working in or near the region of dangerous cavitation for which special sections are required, an upper limiting line for merchant ship propellers of all types, and finally a somewhat lower limiting line suitable for merchant ship propellers with aerofoil type sections.

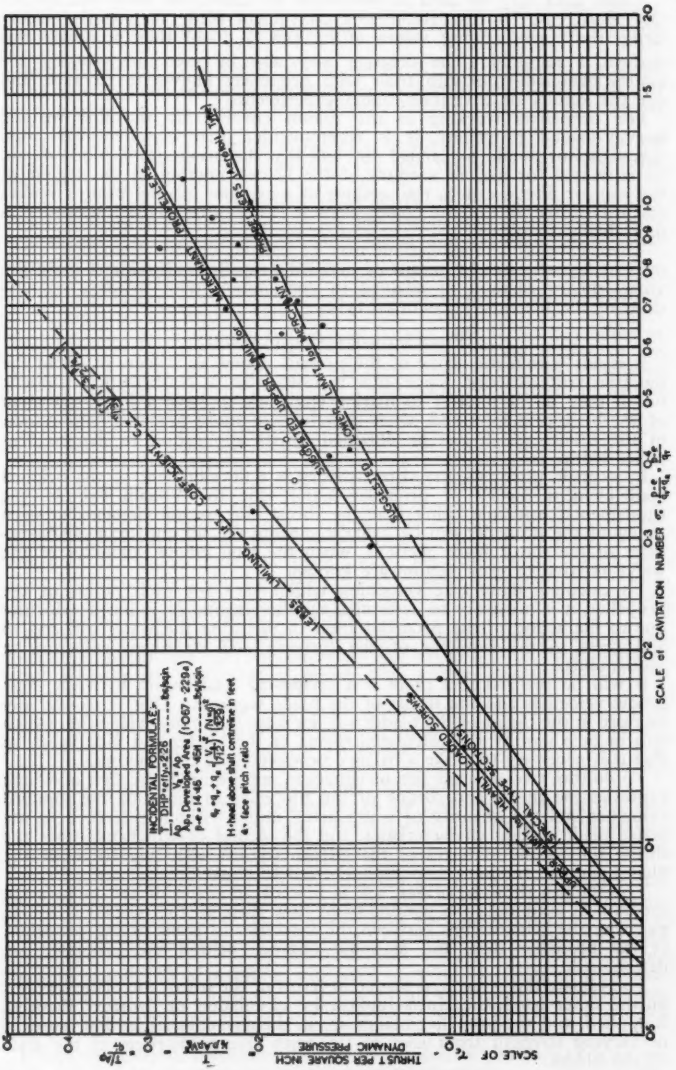

INCIDENTAL FORMULAE:-
DHP=efficiency 225
Density of Area (1057 - 229a)
H=Head above shaft centreline in feet
a = face pitch - ratio
SCALE OF CAVITATION NUMBER
THRUST PER SQUARE INCH DYNAMIC PRESSURE
SCALE OF

FIGURE 9.—CAVITATION DIAGRAM.

These lines may be used to determine the minimum blade surface for any given case by first assuming a convenient blade surface and calculating the corresponding values of σ_0 and τ_{c0}, and then adjusting this surface if required. The diagram may also be used for plotting the results obtained from ship trials with propellers known to cavitate, and for the results of cavitation-tunnel experiments.

For propellers working in the region covered by the line marked heavily-loaded propellers, it is recommended that complete cavitation investigations should be carried out, and that the maximum suctions should be estimated for various sections of the blades, notably in the region of the root or fillet sections, the sections of maximum loading near ·7 radius, and the tip sections of the blades, since it is here that cavitation erosion is most likely to occur.

These cavitation investigations may be carried out by examining specimen or "type" cases by means of detailed propeller calculations using the vortex-theory method of analysis in conjunction with pressure calculations of the Garrick-Theodorsen type.

Vortex-theory Calculations.

It is not within the scope of the present review to discuss in detail the modern vortex-theory, since this is essentially a mathematical subject, but it may be of interest to marine engineers to have a brief statement of the basic principles and, what is perhaps of more practical importance, of the results which can be obtained by its application to marine propeller work.

In its main essentials, the vortex-theory is a development of the original blade-element theory of propeller action due to William Froude, the fundamental principle being that the thrust and torque elements at each radius of the propeller can be calculated from the lift and drag characteristics of the actual blade sections, when the true inflow velocities of the water relative to each section are known.

To calculate these true velocities, which vary for each radius of the propeller, it is necessary to determine the incremental or induced velocities due to the action of the propeller itself in thrusting the water aft, and to add these to the resultant of the forward and rotational velocities at each radius.

It will be realized that there is a reciprocal relationship between these inflow velocities and the thrust of the blade sections. That is to say, any increase in the thrust of the blades will increase the inflow velocities, which will in turn modify the calculated thrust of the blades and vice versa, so that it is necessary to strike a balance between these two factors.

To do this, two separate sets of equations are required, the first of which expresses the thrust and torque of the propeller in terms of the induced velocities and the quantity of water acted upon, without reference to the actual blade section characteristics, and the second of which expresses the thrust and torque in terms of the elementary lift and drag forces at the blades and the resultant velocities referred to above.

To establish these equations, use is first of all made of the momentum theorem in examining the continuity of flow through the propeller, and in this way the relationships between the velocities in front of the propeller, at the propeller disc, and at some distance behind the propeller are determined.

To evaluate the thrust and torque in terms of the lift of the blade sections on the other hand, the Kutta-Joukowsky circulation or "vortex" theorem is applied, and the propeller blades are in theory replaced by line vortices of varying strength from root to tip, with free tip-vortices at the ends of the blades.

This theorem states that the lifting force produced by an aerofoil is

directly proportional to the product of the strength of the circulation flow and the velocity of advance through the medium, so that in this way a relationship is established between the lifting forces and the induced velocities.

By equating the two sets of thrust equations thus obtained, the value of the inflow velocities can be calculated, and it is then possible to determine the elementary thrust, torque and efficiency corresponding to each radius of the propeller.

Strictly speaking, the above explanation applies to the case where the circulation is uniform over the disc, and the propeller has an infinite number of blades.

For actual propellers, allowance must be made for the fact that the blades are small in number, and also for the interference of the blades with each other (i.e., cascade effect), this latter consideration being most important in way of the root sections.

With a finite number of blades, the mean circulation round the elementary annulus under consideration is less than that in the immediate vicinity of the more or less widely separated blades. Corrections for this effect may be applied by employing the original Prandtl equation which refers to the mean circulation for a series of parallel finite lifting planes, or by means of the later determined Goldstein Corrections, which are calculated from an examination of the circulation distribution for a system of helical vortex sheets of appropriate pitch and diameter.

With regard to the corrections for blade interference, these are best determined by applying the results of model tests with series or "cascades" of aerofoils, such as have been carried out by F. Gutsche.

After appyling these corrections in a suitable manner, the final thrust, torque and efficiency at each radius will be known, and these may be integrated to give the total thrust, torque and efficiency for the propeller under consideration. The results of such calculations agree within 5 per cent with the values obtained from model experiments for thrust and delivered horsepower, the calculated figures being approximately 4 to 6 per cent greater than the corresponding test figures, and the efficiencies will be found to agree within 1 per cent.

Applying this type of calculation to the analysis of actual propellers, the distribution of loading over the surface of the blades will be determined in detail, and the lift-coefficient at which each section is working will be known.

It is therefore possible to carry out pressure-calculations by means of the Garrick-Theodorsen method of calculation, published in several of the Reports of the National Advisory Committee for Aeronautics and also referred to in Dr. Lockwood Taylor's recent paper read before the North-East Coast Institution of Engineers and Shipbuilders.

This will enable a complete diagram to be prepared, showing the distribution of pressures and suctions over the face and back of the blades, such as that shown in Figure 10, which refers to a typical heavily-loaded propeller working within the cavitation range.

In a similar manner, diagrams showing the distribution of pressure and suction over the face and back of representative blade sections of different types, such as those shown in Figure 11, can also be obtained.

This comparison shows the pressure distributions for "aerofoil" and segmental sections of the same thickness-ratio, and it will be seen that for the same conditions of working, expressed in terms of angles of incidence relative to the no-lift line or in terms of lift-coefficient, the maximum suction on the back of the former sections is about 30 per cent greater than for the latter type, for the thickness ratio considered.

It will be realized, therefore, that when sections of the former type are to be adopted in any given case, care must be exercised to ensure that the

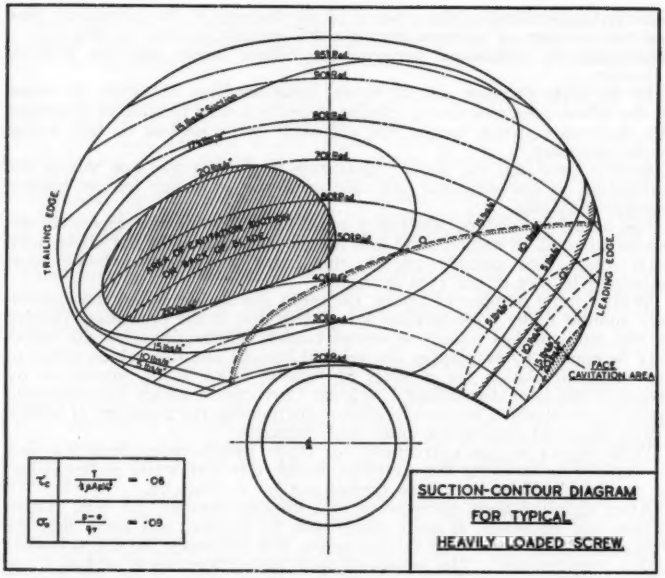


FIGURE 10.

conditions of working are such that local "cavitation" breakdown, or instability of flow, is not likely to occur.

The Vortex-Theory method of analysis can also be used for determining the influence of pitch-variation and the effect of various distributions of wake speed over the disc on the performance and efficiency of any given propeller.

In view of the lengthy calculations involved, this method of analysis is mainly suitable for carrying out "specimen" calculations from time to time when special problems arise, and the data thus accumulated should be regarded in the same way as the results of individual model tests (i.e., as an additional source of information for guidance in preparing subsequent designs).

DEFINITION AND MEASUREMENT OF PROPELLER BLADES.

In order that the best results may be achieved developments in design must go hand in hand with improved methods of manufacture, and the substitution of special "aerofoil" type sections in place of the earlier round-back types for merchant ship propellers, together with the departure from uniform pitch as a necessary essential in propeller design, has thrown into prominence the need for proper definition of proleller blades in a manner most convenient for manufacturing purposes.

For the simpler round-back type of propeller, it was sufficient to define the blade-sections by stating the maximum thickness at each section and indicating that the back of the blades should be formed by circular arcs.

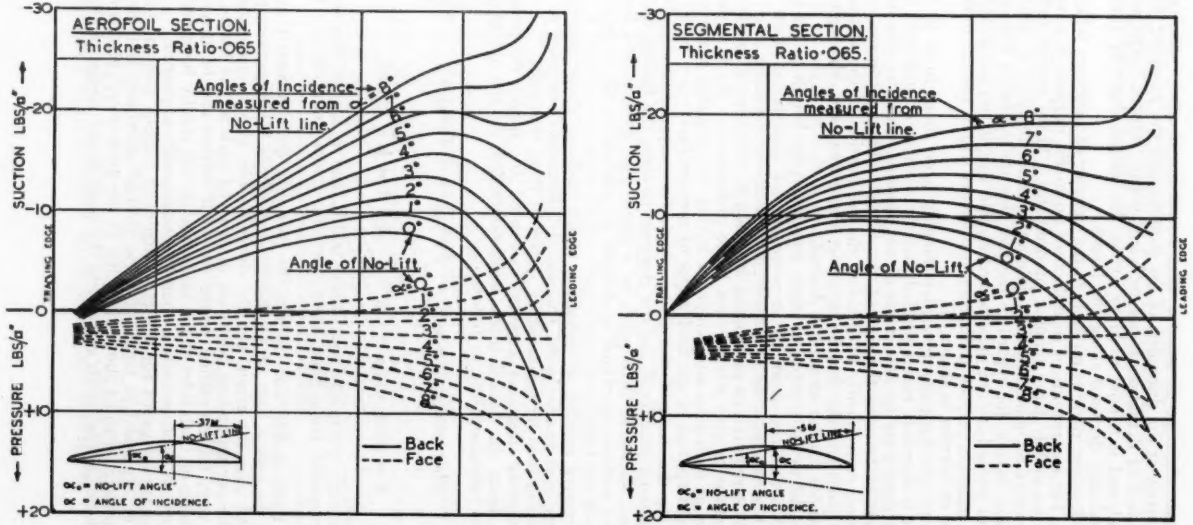


FIGURE 11.—PRESSURE DISTRIBUTION FOR AEROFOIL AND SEGMENTAL SECTIONS.

In some cases, the edge thicknesses were stated, but in many instances this was left entirely at the discretion of the moulder and the propeller finisher.

This method of definition is not sufficient when "aerofoil" type sections are adopted, and it is now usual to define the section shape by means of ordinates suitably disposed along the length.

For the designer, there is no limit to the number of ordinates which can be defined, but for reasons of manufacture no useful purpose is served by employing an unduly large number of ordinates. These section ordinates should therefore be limited to the minimum number required to define the section shape, having due regard to the fact that the propeller finisher will obtain a fair shape between these ordinates, and that if too many ordinates are given, attention may be paid to measurements within normal tolerances rather than to fairness in shape.

For normal merchant-ship propellers ranging from about 10 to 20 feet in diameter it is therefore recommended that the number of ordinates should in no case exceed six in the length of any section, and that this number can be reduced towards the tips of the blades. These ordinates can be obtained most conveniently by dividing the tail portion of the section into four equal parts and the shorter nose part into three parts, although other similar methods can be adopted if desired.

With regard to the number of blade sections in the length of the blades, this should not exceed about six or seven, one of these being given at the crown of boss and another at a short distance above the fillets.

Where it is required to pay special attention to the shape of blade edges, it is convenient to give intermediate blade ordinates for the first three or four inches at each side, and templates having the required shape will be prepared and applied in the final stages of manufacture.

In defining the shape of the blade outline for purposes of manufacture, the conventional practice is for the designer to state the helical widths at each section, measured in relation to a single axis, or directrix line, the longitudinal position of which is defined in relation to the forward and after ends of the boss.

An alternative method of definition which leads to closer precision in manufacture is to determine the true chordal distances from point to point, since these can then be measured by means of fixed trammels.

These chordal distances can be calculated by means of coordinate geometry, from the helical blade widths required by the designer, the appropriate formulae being as follows:

$$\text{Chordal distance} = C = \sqrt{w^2 \sin^2 \theta + 4\, r^2 \sin^2 \beta}$$

where

$$\theta = \text{pitch angle} = \tan^{-1} \frac{p}{2\pi r} \text{ and } \beta = \tfrac{1}{2} \cdot \frac{w}{r} \cdot \cos \theta,$$

where p=pitch, r=radius, w=helical width, and a special diagram can then be prepared embodying these alternative measurements for use in the shops.

Similar methods can also be used for defining the thickness measuring points on the helical face of the blades, and where blade sections parallel to the shaft axis are given, chordal distances can also be calculated for defining the corresponding points on the curved back of the blades.

In this way, the measuring points for each section can be marked off with precision on both sides of the blade, and as will readily be appreciated, the accuracy of measurement is thereby considerably improved.

It is also advantageous for the pitch-angles at various radii of the propeller to be expressed in terms of these chordal distances, and measured on a pitchometer span of about 12 inches in the usual way. Near the root sections these chordal pitch angles and the geometrical design pitch angles

vary quite considerably, and errors in pitch can occur if the latter angles are used in conjunction with a fixed pitchometer span, such as frequently occurs when this point is not fully appreciated.

Machining of Modern Varying-pitch Propellers.

In recent years it has been the practice for all important propellers to be planed to pitch by means of suitable pitch-planing machines. This method can also be extended with advantage to the planing of varying-pitch propellers, provided a regular curved pitch-variation such as can be represented by the formula:

$$\text{pitch} \propto \tan^{-1}\left[\frac{\text{r. sin } \beta}{\text{A} + \text{r. cos } \beta}\right]$$

(where A=constant, β=included angle of the projected blades, r=radius of section under consideration), is adopted by the designer, in the initial stages.

Figure 12 shows the type of pitch-reduction lines thus obtained, and it will be noted that these correspond very closely with the pitch-reduction lines adopted by many designers in connection with aerofoil type propellers of modern design. The diagram on the left-hand side of Figure 12 may be used to determine the pitch reduction curve for any given case when the mean face pitch and the pitch reduction at ·2R have been decided.

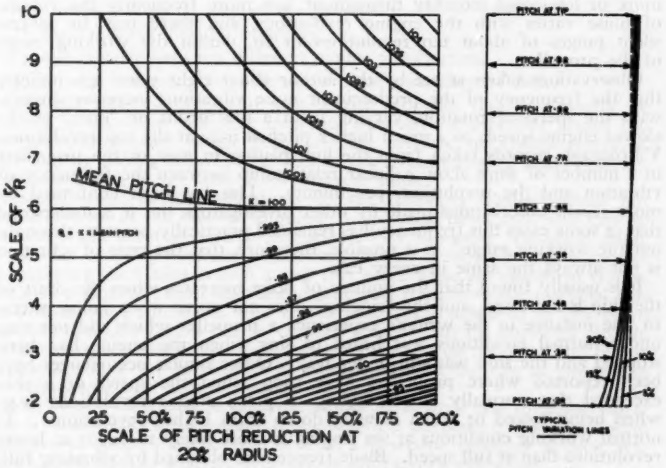


Figure 12.—Pitch-reduction Diagram.

The appropriate pitch-bed for propellers having a pitch-reduction line of this type can also be readily swept up in the mould in one operation, thus avoiding the more complicated and difficult methods which must be employed when the pitch variation adopted is irregular.

"Singing" Propellers.

Much attention has in recent years been focused on the question of "singing" propellers. If this had not been so, it seems likely that many of the cases which have been reported would have remained unnoticed, since it is frequently stated by marine engineers, who have been at sea for a long time, that similar noises occurred many years ago without attracting special attention.

At the same time, it appears to be clearly established by recent experience that unless special measures are taken to avoid this phenomenon, there is much greater likelihood of its occurrence with "aerofoil" type propellers, than with the simpler round-back elliptical type of propeller.

So far as the mechanism of "singing" is concerned, it is now generally agreed that the noise is caused by vibration of the propeller blades, and that the sound-vibrations are transmitted through the water and picked up by the shell-plating in way of the after end of the ship, which acts as a sounding board.

In some cases the shaft-tunnel also serves to amplify the sound, and conditions similar to those obtaining in an organ-pipe occur. Due to the cyclical variation in wake conditions round the disc of the propeller, and also in the position of the blades relative to the hull during each revolution, the singing noise usually rises and falls in such a way that the engine revolutions can be counted at any time, thus making the sound more audible to those inside the ship.

Sometimes the singing noise occurs at all revolutions, and may be of more or less equal intensity throughout, but more frequently the volume of noise varies with the engine revolutions, and there may be several silent ranges of about ten revolutions or so, within the working range of the propeller.

Observations taken at sea by the author about eight years ago indicate that the frequency of the predominant noise vibrations increases directly with the speed of rotation, varying from a low grunt or "hum" at the slower engine speeds to a much higher pitched note at the top revolutions. Vibrograph records taken from the hull plating in way of the propellers in a number of ships show a linear relationship between the frequency of vibration and the revolutions per minute. This has been confirmed by more recent observations made by other investigators, but it is understood that in some cases this frequency has remained practically constant throughout the working range. It is possible, therefore, that the type of vibration is not always the same in every case.

It is usually found that the volume of noise increases when the draft of the ship is increased, and the "singing" does not occur when going astern. In one instance in the writer's experience a propeller which did not sing under normal conditions was heard to sing when the engine had been stopped and the ship was still under way. Other similar occurrences have been reported where propellers have sung when the speed of a ship exceeded that normally corresponding to given engine revolutions (e.g., when being towed or when slowing down from higher revolutions). In normal working conditions at sea singing is more likely to occur at lower revolutions than at full speed. Blade frequencies obtained by vibrating full-size propellers in air agree reasonably well with the corresponding "singing" frequencies, and from the evidence of such tests it appears that for merchant-ship propellers the vibrations concerned are mainly of a torsio-flexural type, in which the natural flexural vibrations of the blades are accompanied by a torsional movement of the tip sections, although the double-torsional vibrations recently suggested by Dr. Lockwood Taylor may offer an alternative explanation of the kind of motion which is present.

The linear relationship mentioned above suggests that when singing occurs the pitch of the note may be governed by the frequency of eddy-

shedding corresponding to the speed of the outer parts of the blades. Furthermore the louder ranges of revolutions may correspond to conditions when an approach to synchronism is present, with respect to a natural mode of vibration. It may also happen that in some instances the frequencies corresponding to different modes of torsional and flexural vibration are approximately in agreement, and that complex vibrations involving more than one mode may account for the harsh unmusical sounds sometimes heard, when occasioned by the above circumstances. It is not suggested that the eddy-shedding or "aeolian harp" effect is the prime cause of singing vibrations, since this is present with all propellers, but rather that the alternate shedding of eddies in a regular manner may influence the pressure distribution round the sections and thus control the frequency in cases where instability is likely to occur due to other causes.

Finally, where the conditions of flow in way of a propeller are extremely variable, due to complicated wake patterns, constricted apertures, bluff rudder shapes, etc., it is likely that the "buffeting" effect which is present in these circumstances may cause the blades to be continually in vibration, so that "singing" is difficult to avoid.

From the foregoing general observations, it will be appreciated that although the precise nature of the mechanism of "singing" is still unknown, and calculations can not be made to determine whether or at what revolutions a given propeller will sing, practical measures may be taken to avoid all the possible causes of blade-vibration enumerated.

These causes may be summarized as follows:

(1) Unsteady flow in way of the propeller,

(2) natural torsio-flexural coupling,

(3) interference between different modes of vibration,

(4) rapid pressure-gradients liable to cause instability of flow.

Unsteady flow conditions in way of the propeller disc can usually be avoided by providing sufficient clearance between the propeller blades and the surrounding parts of the hull (i.e., fins, rudders, sternframe-apertures, etc.) and by paying careful attention to the streamlining of shaft-bossings, A-brackets, etc., in the case of twin screw ships.

Natural torsio-flexural vibrations of the propeller blades should not normally arise in the case of elliptical round-back propellers, and can be avoided when aerofoil sections are adopted by adjusting the blade shape so that the coupling between flexure and torsion is reduced—that is to say, by designing the blades in such a way that when these are caused to vibrate freely in a flexural mode, unbalanced moments relative to the torsional axis are reduced to a minimum. To ensure that this is achieved in any given case, the propeller blades can be vibrated freely in air by fitting an electric motor provided with a small out-of-balance weight on the after end of the boss, and noting the amplitudes of vibration round the tips of the blades at various speeds of rotation of the motor.

Interference between the different modes of flexural and torsional vibration is more difficult to avoid, but for practical purposes it has been found sufficient to adopt a distribution of mass and section modulus in each new design, based on an analysis of previous satisfactory propellers.

So far as the final condition mentioned above is concerned, this can be dealt with by the provision of sufficient blade surface for the working conditions of the propeller, and by avoiding blade-sections having unduly high local suction-peaks. This is of course most important in way of the outer sections of the blades, where the natural circulation is modified by the action of the tip vortices, and special care must therefore be taken in this region.

It is realized that these practical measures to avoid the occurrence of blade vibration do not represent a complete solution of the singing problem,

but experience over a number of years has shown that singing is no longer to be feared when these principles are adopted.

Another expedient which has been suggested for the purpose of reducing singing has been the adoption of very fine blade edges without altering the general design of the propeller. When this is done, the hydrodynamic damping is increased due to the high impedance to flow round the sharp edges when lateral motion occurs. In some cases, an alteration of this nature has proved successful in reducing or eliminating the singing noises, but in others this expedient has had little success. In one instance, in fact, the singing increased after the leading and trailing edges had been sharpened in this way, and the propeller was later silenced by an adjustment in blade thickness, which included cutting back the edges to give a more normal edge thickness.

From the point of view of handling the propellers and of extreme liability to damage in service, very sharp edges are not favored by marine engineers, and since propellers having normal edge thicknesses can be adopted without the occurrence of singing, it is recommended that this type of modification should only be adopted in special cases or as a temporary measure.

Variable-pitch Propellers.

Up to the present time, the application of the variable-pitch principle to marine screw propellers has been limited to propellers up to about 5 feet in diameter fitted to lake-ferries and other small craft, but quite recently one or two larger propellers of the Kamewa type have been made in Sweden for the purpose of trials on larger sea-going ships. In view of this, it is thought that some brief notes on the possible design advantages to be gained by the adoption of variable-pitch propellers for merchant ships will be of interest to marine engineers.

The novel feature of the variable-pitch screw for marine work lies of course in the fact that the propeller revolutions can be varied within very wide limits for any given power or ship speed, by adjusting the pitch of the blades, whereas in the case of a normal installation with fixed-pitch propeller, the engine revolutions for any ship speed are automatically controlled by the propeller, which acts as a dynamometer or hydraulic brake.

For most ships of normal form it is found that the engine revolutions for different ship speeds are almost directly proportional to speed, so that the propeller slip remains practically constant over a wide range of speed, and the power varies approximately as the cube of the revolutions. At first sight, this may be thought to restrict the engine and propeller revolutions in an unfavorable manner, but in actual fact, from the point of view of propeller efficiency at different speeds, the relationship is such that the propeller works under more or less constant conditions. For example, taking the case where the Admiralty coefficient remains approximately constant over the working range of speeds, we have

$$\frac{\Delta^{\frac{2}{3}}V^3}{\text{I.H.P.}} = \text{Constant.}$$

and assuming constant displacement and IHP $\propto N^3$ it will be seen that

$$\frac{V^3}{N^3} = \text{Constant or } \frac{V}{N} = \text{Constant,}$$

so that the basic variable $\delta = \frac{Nd}{Va}$ also remains constant.

In the same way $\frac{N^3}{IHP}$ is constant so that A_Q which is proportional to

$\frac{P}{N^3}$ also remains constant, and therefore the position at which the propeller is working in the Design-Analysis diagram shown in Figure 5 remains the same throughout. As a result of this, the propeller designed for optimum efficiency at full power in mean weather conditions will maintain this efficiency over a wide range of speed. This is confirmed by the analysis of voyage records for normal merchant ship types, from which it is found that under average weather conditions at sea the value of the basic variable B_p remains practically constant at different ship speeds.

Owing to the effects of heavy weather and fouling, etc., the hull resistance will of course be increased at various times, and this will be reflected in the power required for any desired speed. On the other hand, in specially favorable weather conditions or with decreased draft, the hull resistance will be decreased. In order to examine what happens in respect of propeller efficiency under these conditions, a typical example has been worked out for a merchant ship fitted alternatively with a fixed-pitch propeller and a variable-pitch screw designed for 3640 D.H.P. at 110·4 Rpm., with a ship speed of 13·0 knots, and the results are shown in Tables I and II.

VARIABLE PITCH PROPELLERS.

EXAMINATION OF PROPELLER EFFICIENCY CHANGES.

BASIC DESIGN CONDITION: 13·0 KNOTS. 110 RPM. 3640 D.H.P.

TABLE I.

(1) EXAMINE LOWER SPEEDS IN GOOD WEATHER.

		Fixed.		*Variable.*		*Improve-*
D.H.P.	*Speed.*	*Rpm.*	*Effcy.*	*Rpm.*	*Effcy.*	*ment.*
2840	12·0	102	·565	86	·576	1·8%
1634	10·0	85	·565	49·5	·530	–6·0%
1634	10·0	—	—	67	·576	1·8%
	optimum					
3640	13·0	110	·565	110	·565	Nil

TABLE II.

(2) VARIATIONS IN WEATHER AND LOADING FOR 12 KNOTS.

		Fixed.		*Variable.*		*Improve-*
D.H.P.	*Condition.*	*Rpm.*	*Effcy.*	*Rpm.*	*Effcy.*	*ment.*
1980	—30% E.H.P.	92·6	·584	75·0	·612	4·9%
2272	—20% E.H.P.	95·8	·579	78·2	·602	4·0%
2556	—10% E.H.P.	99·5	·571	82·0	·590	3·3%
2840	Normal Sea	102	·565	86·0	·576	1·8%
3124	+10% E.H.P.	105	·557	95	·562	0·8%
3408	+20% E.H.P.	107·8	·551	103	·554	0·5%
3690	+30% E.H.P.	110	·545	110	·545	Nil
	(3) 13½ KNOTS FOR 2840 D.H.P. (E.H.P.—30%).					
2840	—30% E.H.P.	104·6	·583	102	·589	1%

N.B.—The above figures do not include any allowance for initial loss in efficiency due to adoption of larger boss, etc., for variable pitch propeller. This loss would be at least 6 to 7 per cent with most favorable conditions.

Table I shows the revolutions and propeller efficiencies obtainable at 12 knots and 10 knots in good weather. In the case of the fixed pitch screw the Rpm. are automatically reduced to 102 Rpm. and 85 Rpm. respectively, and the propeller efficiency remains approximately constant. With the variable-pitch screw on the other hand the revolutions can be reduced to

86 and 49·5 respectively without exceeding the design limits of mean effective pressure. In the first instance there is a gain in efficiency of 1·8 per cent and in the second case a loss of 6 per cent, but in this latter case the Rpm. can be increased to 67 Rpm., and the gain of 1·8 per cent can be obtained in this way.

Considering the 12 knot condition in more detail, and assuming the E.H.P. to vary between 70 per cent and 130 per cent of normal sea conditions, to represent lighter loading and heavy weather respectively, the optimum conditions obtainable with a variable-pitch propeller are shown in Table II. From this table, it will be seen that for the light load conditions gains up to 5 per cent can be obtained, but for the heavy weather or fouling conditions the gains are progressively smaller, as the resistance is increased.

In obtaining these figures it has been assumed that the variable-pitch propeller can be made to be of equal efficiency to a fixed pitch screw for similar conditions of working, but this is, of course, not true. In actual fact, owing to the larger boss required, which must at least equal that of a loose-bladed or "built" propeller (and will no doubt be larger owing to the pitch changing mechanism which has to be accommodated), and due also to the narrow, thick root sections which must be adopted, the efficiency of the variable-pitch propeller will probably be at least 6 per cent or 7 per cent less than that of the equivalent fixed-pitch screw.

When due allowance is made for this, it will be seen that under the most favorable conditions the variable-pitch screw will be 2 per cent less efficient than the corresponding fixed-pitch screw, and under normal working conditions the loss will be about 6 per cent to 7 per cent.

Since the governing condition in designing a new propeller is usually the mean effective pressure which is obtainable at full-power revolutions, the power developed at lower revolutions is less than that which could be obtained if a variable-pitch screw were fitted, and the mean effective pressure falls off rapidly at lower speeds. This is accentuated in light-load conditions, and in order to determine the effect on propulsive efficiency the −30 per cent E.H.P. condition at the head of Table II has been examined further as follows:−

It will be seen that the D.H.P. for 12 knots in this condition is 1980 as compared with the normal power of 2840 D.H.P. for this speed. It is interesting therefore to examine what happens in this case, if it is decided to use the full 12-knot horsepower. By calculation, it is found that the revolutions increase to 104·6 Rpm. for a ship speed of 13½ knots with 2840 D.H.P. The propeller efficiency with fixed pitch rises from ·565 to ·583 owing to the change in slip conditions, and if by varying the pitch the revolutions are reduced to 102 Rpm. (i.e., the normal Rpm. for this power) a further gain in efficiency of 1 per cent can be obtained.

From the foregoing, it will be seen that for these very wide differences in ship conditions, the advantage in efficiency to be gained from fitting a variable-pitch screw is very small, and when full account is taken of the losses due to boss size, etc., it is evident that, from the point of view of propeller efficiency, the variable pitch screw does not appear to offer any worth while gains for merchant ships of normal type.

The reason for this lies in the fact that the diameter of the propeller is fixed, and cannot be varied at the same time as the pitch, so that possible changes in position on the Design-Analysis diagram are, in any given case, restricted to variations along a single line, which lies approximately parallel to the efficiency contours shown in Figure 5. Similar calculations have been made for widely varying types of merchant ship screws with the same result. It remains, therefore, to be seen whether the extreme flexibility in engine revolutions will be found by marine engineers to offer any material gains in fuel economy or in respect of engine wear which would

make the considerable extra cost and upkeep of such an installation worth while in any given case.

In special types of ships such as tugs and very fast speed boats, where the variations in resistance with speed are abnormal, owing to different sizes of tow in the one case and due to planing in the other, the circumstances are quite different from those outlined above, and in these cases material gains in power absorption can be achieved with the variable-pitch screw since the pitch which is appropriate to full power conditions may limit the power which can be developed at lower speeds. This is specially true when fast running Diesel or boosted petrol engines are employed.

Finally, for river or inland water craft, the improved maneuvering conditions obtainable with variable-pitch screws will be extremely attractive, and the ability to go ahead or astern or to feather the blades without adjusting the engine setting is very valuable in such vessels.

In the above brief review, only the main design considerations in relation to propeller efficiency have been outlined, and there is room for considerable discussion of the practical implications involved in the adoption of variable-pitch screws, such as reliability, wear, damage, replacements, control mechanism, etc., and when the initial mechanical difficulties have been overcome, as there seems no doubt they very soon will be, these practical questions will be the subject of many lively debates and discussions among marine engineers.

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MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

EXPERIMENTAL FLUID DYNAMICS APPLIED TO ENGINEERING PRACTICE.—In this lecture, G. A. Hankins, Principal Scientfic Officer of the Engineering Department, National Physical Laboratory, illustrated the many valuable practical applications of experimental fluid dynamics in nearly all branches of engineering. Dynamical similarity in fluid motion and the use of the Reynolds, Froude and Mach numbers in laboratory experiments are first considered. Mr. Hankins then describes several practical applications of the basic principles. His lecture was delivered before a general meeting of the North-East Coast Institution of Engineers and Shipbuilders on October 29, 1943.

In addressing you on fluid dynamics applied to engineering practice, it cannot be suggested that I have selected a subject which is limited in scope or lacking in importance to engineers and shipbuilders. By definition, dynamics is that branch of science which deals with the action of forces in producing or affecting motion, so that fluid dynamics is a fundamental science which finds applications in nearly all branches of engineering. Despite this, the subject as now understood is of comparatively recent growth, and some of you will doubtless remember that thirty years ago it consisted mainly of hydrodynamics, which was mathematically exact but had little practical application, and hydraulics, which was essentially practical but highly empirical. It is true that there were useful applications of hydrodynamic theory to engineering problems, but they were limited by the fact that general solutions of the fundamental mathematical equations of the flow of viscous fluids such as air and water could not be obtained. The remarkable growth of the subject in recent years can be largely attributed to the developments of theoretical and experimental aerodynamics, and the practical results, applied to aircraft design and performance, have been brought very forcibly to our notice during the past four years. As a contributing but less spectacular factor, the knowledge gained from ship tank research is also important. In these developments much progress has been made in the application of solutions of the basic mathematical theory of fluid motion to practical purposes, but of even greater value perhaps, has been the extensive use of laboratory experiments carried out on a logical scientific basis.

The success which has attended the application of wind-tunnel and ship-tank experiment to aircraft design and practical shipbuilding respectively

has resulted in a realization of the usefulness of experimental fluid dynamics in other branches of engineering, and it is some of these more general applications that I propose to consider in this lecture. My main object is to present the subject as a branch of engineering science with which all engineers should be familiar, because it is proving of great and increasing value in many fields of engineering other than aeronautics and naval architecture, and because experience suggests that engineers, in general, do not yet seem to appreciate the full value of the tool now available for their use. Necessarily, I cannot hope to deal with a large number of applications in any detail, but I can endeavor to outline the more important general principles involved, and give some illustrations of engineering interest. The principles are important since they enable the practicing engineer to appreciate the subject as a related whole, and not as a collection of diverse phenomena which at first sight appear to have little in common. Moreover, a knowledge of principles permits the engineer to distinguish for himself between fundamentally sound experimental data which can be expected to give reliable information concerning a proposed new design, and the more speculative experiment in which, for some reason or other, it has been necessary to depart from the established principles. The illustrations are not intended to provide an exact statement of the present state of knowledge in the particular spheres with which they deal, but are meant to give an indication of the way in which laboratory experiment is being applied to engineering matters, which, until a few years ago could only be considered in the light of previous practical experience. With one exception, the subjects selected for the illustrations have all been investigated in some detail in the Engineering Department of the National Physical Laboratory.

Dynamical Similarity and Dimensional Analysis.

In dealing with experimental problems in fluid dynamics the basic principles of dynamical similarity and dimensional analysis are of paramount importance since it is by the application of these principles that experiments can be made on scale models and the results transposed to full-scale prototypes; moreover by the use of these same principles experiments can be carried out with one fluid and the results applied to a variety of other fluids. The general idea of a scale model geometrically similar to the full-scale prototype is easily visualized, and involves only one ratio, namely that of the linear dimensions. But for model experiments in fluid dynamics to be of the greatest value it is highly desirable for the fluid motions in the two systems to be dynamically similar; that is to say, the geometrical similarity has to apply not only to the shape of the bodies, but also to the paths of the motion of corresponding points in the two fluid systems. For example, if an eddy or wave occurs in the full-scale system, a geometrically similar eddy or wave should be present in the model system. To obtain this the fluid forces must be such that the relative positions of the parts of the fluid in one system after a time t_1 must be geometrically similar to those in the other system after a corresponding time t_2, and the ratio of t_1 to t_2 must be constant. For the fluid forces to produce this result they must preserve the same ratios to one another at corresponding points.

The main forces in the fluid with which we are concerned are (1) inertia forces due to the mass of the fluid, (2) forces due to the viscosity, (3) gravitational forces, (4) compressibility or elastic forces. If inertia forces only are concerned the requirement for dynamical similarity is simply that the forces be proportional to the mass multiplied by the acceleration, but in a very large number of cases which arise in practice, both the inertia forces and forces due to the viscosity of the fluid are highly

important. Dynamical similarity for two systems then requires that the ratio of these two forces should be constant; this requirement can be written in the form

$$\frac{\text{Inertia force}}{\text{Viscosity force}} \equiv \frac{vl}{\mu/\rho} \text{ (Reynolds number)} = \text{constant}$$

where v is the velocity, l a typical linear dimension of the system, μ the viscosity,* and ρ the mass density. Thus in comparing the behavior of two geometrically similar bodies or systems, under the combined action of inertia and viscosity forces the requirement for correct experiment is that the Reynolds number should be the same in both. This is the principle on which wind-tunnel experiments in aerodynamics are based, and by which flow in pipes can be compared.

Gravitational forces necessarily arise in fluid motion when the free surface of a liquid is displaced above or below a mean level in flow between solid boundaries or by the movement of a body such as a ship on the surface of the liquid. The essential requirement for dynamical similarity in two systems in this case is a constant ratio between the inertia forces and the gravitational forces. This requirement can be written in the form

$$\frac{\text{Inertia force}}{\text{Gravitational force}} \equiv \frac{v^2}{lw/\rho} = \frac{v^2}{lg} = \text{constant}$$

where w is the weight density ($w/\rho = g$ the acceleration due to gravity). The parameter $\frac{v}{\sqrt{lg}}$ is the Froude number. The constant Froude number is, of course, the basic principle on which ship-model tests are carried out and it is also used in models of the flow of water over weirs and spillways.

When consideration is given to fluid motion in which the velocities approach or exceed the velocity of sound in the fluid, the effect of the elastic forces in the fluid becomes highly important. For adiabatic changes the modulus of elasticity of a gas E is equal to $\gamma\rho$, where ρ is the pressure and γ is the familiar ratio of specific heat at constant pressure to that at constant volume. Dynamical similarity in two corresponding systems then requires a constant ratio between the inertia forces and the elasticity forces. This requirement can be written in the form

$$\frac{\text{Inertia force}}{\text{Elasticity force}} \equiv \frac{v^2}{E/\rho} = \frac{v^2}{a^2} = \text{constant}$$

where E is the modulus of elasticity and $\sqrt{\frac{E}{\rho}} = a$ the velocity of sound in the fluid. The parameter v/a is usually known as the Mach number, and in making experiments on model projectiles it is necessary that the Mach number should be the same for the model experiment and the full scale.

The basic principle of dimensional analysis is that all the terms of a correct physical equation must be dimensionally homogeneous, from which

* Viscosity is defined as the tangential force per unit area per unit rate of shear on adjacent layers of a moving fluid, i.e.,

$$\text{Tangential stress} = \mu\frac{dv}{dy} \text{ where } \frac{dv}{dy} \text{ is the velocity gradient.}$$

it follows that if we have a relationship in fluid dynamics connecting certain dimensions, the two sides of the equation when expressed in terms of the three fundamental dimensions of length, time and mass, must give identical powers of these three dimensions. A common illustration is that of the flow of a fluid through a long uniform horizontal pipe at a distance sufficiently far from the entrance for uniform flow conditions to have been reached. We may perhaps expect the pressure gradient, that is, the pressure loss in the fluid per unit length of pipe, to be dependent only on the pipe diameter d, the mean flow velocity v, the mass density ρ and the viscosity μ of the fluid. This expectation can be written in the form of an equation

$$\frac{\delta p}{\delta l} \text{ (the pressure gradient)} = \phi\,(d^w\, v^x\, \rho^y\, \mu^z) \quad . \quad . \quad . \quad (1)$$

where ϕ denotes an unknown function of the quantities inside the bracket. Dimensionally the two sides of equation (1) must agree and on this basis it can easily be shown that

$$\frac{\delta p}{\delta l} \Big/ \frac{\rho v^2}{2d} = \phi_1\left(\frac{vd\rho}{\mu}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

This is, of course, a well-known text book example, and $\frac{vd\rho}{\mu}$ is the Reynolds number for pipe flow. The point I wish to emphasize is that the dimensional derivation is in itself no proof that the resulting relationship is correct unless the original assumptions are sound. Such proof and the form of the unknown function $\phi_1\left(\frac{vd\rho}{\mu}\right)$ must be obtained by other means. The proof of the validity of Equation (2) for flow in smooth pipes was first obtained in the well-known experiments of Stanton and Pannell (1)* carried out at the National Physical Laboratory in 1914. They used water, air and oil, and by plotting values corresponding to the left-hand side of Equation (2) against $\frac{vd\rho}{\mu}$ showed that all the results fell on a continuous curve irrespective of the individual variations in v, d, ρ and μ, and thus verified the original assumptions. In the case of rough pipes, the results do not fall on a continuous curve when the Reynolds number is above a particular critical value, thus showing that for such conditions the original assumptions are not correct. The value of $\phi_1\left(\frac{vd\rho}{\mu}\right)$ of Equation (2) is usually known as the friction coefficient and denoted by λ, and is dependent only on the Reynolds number for the flow of any incompressible fluid in a smooth horizontal pipe. When the pipe is not smooth, λ is a function not only of the Reynolds number, but also of one or more dimensionless parameters representing the surface roughness.

An important application of the principles of dynamical similarity and dimensional analysis covering a wide range of experimental fluid dynamics is the determination by model experiments of the forces acting on a body moving in a straight line through a fluid at constant speed. Considering the resistance D directly opposing the motion, the general case, in which fluid forces due to inertia, viscosity, gravity, and elasticity are all assumed to act, can be expressed by the dimensionally correct relationship.

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* See list of references p. 159.

$$\frac{D}{\frac{\rho v^2 l^2}{2}} = \phi\left(\text{shape}, \frac{vl}{\mu/\rho}, \frac{v^2}{\frac{l}{\rho}w}, \frac{v^2}{\frac{E}{\rho}}\right)$$

$$= \phi\ (\text{shape, R, F, M}) \quad . \quad . \quad . \quad . \quad (3)$$

where R is the Reynolds number, F is the Froude number, M the Mach number and $\frac{D}{\frac{\rho v^2 l^2}{2}}$ is a non dimensional resistance coefficient.

In actual model experiments it is only practicable to work on the basis of one dimensionless parameter but the relationship shows the underlying connection between model experiments in (*a*) normal wind-tunnel tests, in which F and M are omitted, (*b*) ballistic model tests, in which R and F are omitted, and (*c*) ship tank model tests, in which F is the basic criterion and a correction is then made for R. The object of the model experiments in each case, of course, is to determine experimentally the variation of the resistance coefficient with the particular controlling parameter, R or F or M, and typical results are shown in the three diagrams of Figure 1. Figure 1*a* shows the non-dimensional resistance coefficient of Equation (3) plotted against Reynolds number for a circular disc, a sphere and an airship model, completely immersed in the fluid; the results apply generally to the particular shaped body moving in any fluid in which compressibility effects can be neglected. It will be noted that the resistance of the slender airship shape is less than 1/20 of that of the flat disc of the same cross-sectional area; in the case of the sphere the resistance coefficient varies considerably with Reynolds number. Figure 1*b*† illustrates the difference in wave-making resistance between a slow cargo ship and a high-speed vessel, the residuary resistance (*i.e.* the total resistance minus the friction) being plotted as a resistance coefficient‡ against Froude number. It may be remarked that the curves of Figure 1*b* are not exactly representative of wave-making resistance since the residuary resistance includes eddy resistance which should really be considered on a Reynolds number basis. Figure 1c shows the effect of compressibility, the resistance coefficient in this case being plotted against Mach number; the chief point of interest is the marked increase in the resistance coefficient in the neighborhood of a Mach number of 1, that is, in the region of the velocity of sound in the fluid.

Figure 1 is thus illustrative of the three main divisions of experimental fluid dynamics. There is a further important distinction, however, which must be borne in mind in relation to all fluid flow phenomena in which viscosity forces arise, namely, the difference between viscous (or laminar) flow and turbulent flow. In viscous flow the forces exerted on or by the fluid are directly derived from the tangential or shearing stresses between parallel layers in the moving fluid; at low speeds or with thin films, fluid motion can be considered on this basis. In turbulent flow there are velocity fluctuations in the fluid both along and across the mean flow direction which result in a loss of energy much greater than would arise from viscous flow under similar conditions. An important example of viscous flow in engineering practice is the lubricated bearing, and this illustration will be considered first; consideration will then be given to flow in pipes, in which both viscous and turbulent flow can occur. The subsequent illustrations will deal with wind forces on buildings, river and tidal models, and fluid motion at high speeds.

† I am indebted to Dr. F. H. Todd for the information shown in Figure 1b.

‡ The midship sectional area has been used in Figure 1b to preserve uniformity with Figures 1a and 1c; actually beam is much more important than draft in wave making resistance.

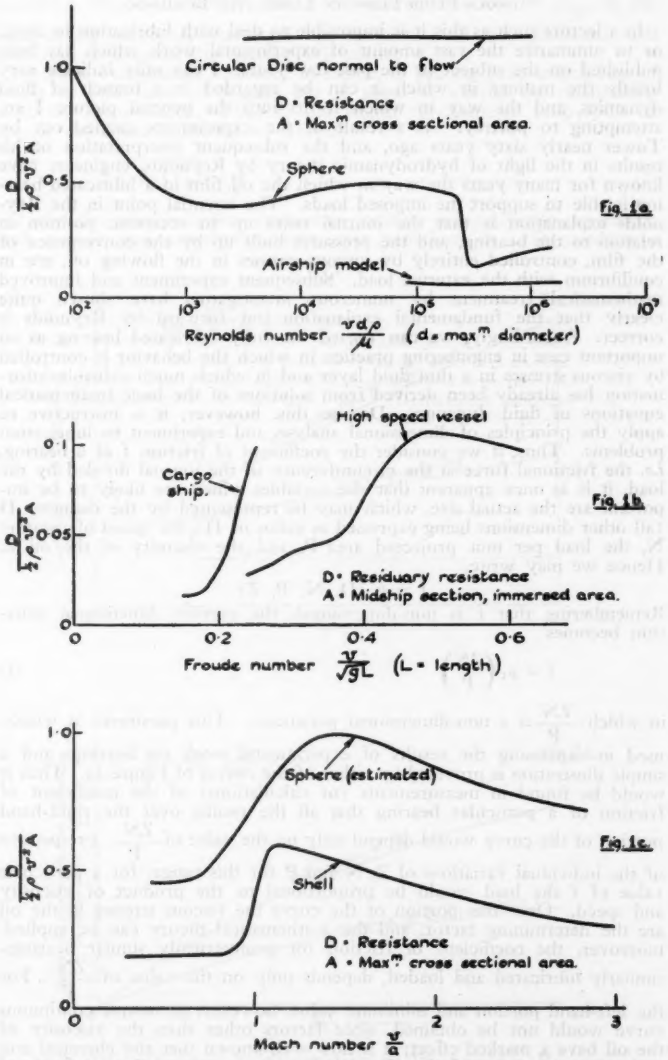

FIGURE 1.—VARIATION OF RESISTANCE WITH REYNOLDS, FROUDE, AND MACH NUMBERS.

Viscous Fluid Flow in Lubricated Bearings.

In a lecture such as this it is impossible to deal with lubrication in detail or to summarize the vast amount of experimental work which has been published on the subject in the past few years. I can only indicate very briefly the manner in which it can be regarded as a branch of fluid dynamics, and the way in which it fits into the general picture I am attempting to portray. As a result of the experiments carried out by Tower nearly sixty years ago, and the subsequent interpretation of his results in the light of hydrodynamic theory by Reynolds, engineers have known for many years the way in which the oil film in a lubricated bearing is able to support the imposed loads. The essential point in the Reynolds explanation is that the journal takes up an eccentric position in relation to the bearing, and the pressures built up by the convergence of the film, controlled entirely by viscous stresses in the flowing oil, are in equilibrium with the external load. Subsequent experiment and improved mathematical treatment by numerous investigators have shown quite clearly that the fundamental explanation put forward by Reynolds is correct. Accordingly, we can regard the fully lubricated bearing as an important case in engineering practice in which the behavior is controlled by viscous stresses in a thin fluid layer and in which much valuable information has already been derived from solutions of the basic mathematical equations of fluid dynamics. Despite this, however, it is instructive to apply the principles of dimensional analysis and experiment to lubrication problems. Thus, if we consider the coefficient of friction f of a bearing, *i.e.* the frictional force at the circumference of the journal divided by the load, it is at once apparent that the variables which are likely to be important are the actual size, which may be represented by the diameter D (all other dimensions being expressed as ratios of D), the speed of rotation N, the load per unit projected area P, and the viscosity of the oil Z. Hence we may write,

$$f = \phi \ (D, N, P, Z)$$

Remembering that f is non-dimensional, the correct dimensional equation becomes

$$f = \phi_1\left(\frac{ZN}{P}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which $\frac{ZN}{P}$ is a non-dimensional parameter. This parameter is widely used in expressing the results of experimental work on bearings and a simple illustration is provided by either of the curves of Figure 2a. Thus it would be found in measurements (or calculations) of the coefficient of friction of a particular bearing that all the results over the right-hand portion of the curve would depend only on the value of $\frac{ZN}{P}$, irrespective of the individual variations of Z, N and P for this range; for a particular value of f the load would be proportional to the product of viscosity and speed. Over this portion of the curve the viscous stresses in the oil are the determining factor, and the mathematical theory can be applied; moreover, the coefficient of friction for geometrically similar bearings, similarly lubricated and loaded, depends only on the value of $\frac{ZN}{P}$. For the left-hand portion and minimum value, however, an unique continuous curve would not be obtained, since factors other than the viscosity of the oil have a marked effect; it is now well-known that the chemical and molecular properties of the oil, and the nature and mechanical finish of the bearing surfaces are important in this region. Normally, the engineer

wishes to design a bearing so that it will operate at values of $\frac{ZN}{P}$ to the right of the minimum since here the complete fluid film conditions will hold and no wear of the bearing will occur.

Equation (4) assumes similarity in regard to bearing proportions, line of action of load, and oil supply, but if these be altered the relationship must be reconsidered. Thus if the length to diameter ratio L/D be altered, the equation should be written

$$f = \phi_2\left(\frac{ZN}{P}, \frac{L}{D}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

and the two dimensionless parameters studied independently. Actually, both theory and experiment show that variations in L/D produce a

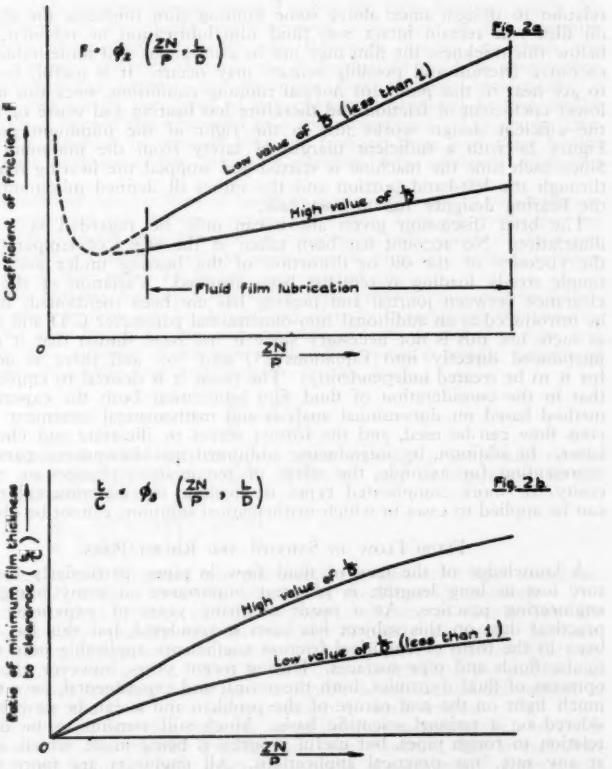

FIGURE 2.—DIAGRAMMATIC REPRESENTATION OF DIMENSIONAL RELATIONSHIPS APPLIED TO LUBRICATION.

family of curves of the type shown in Figure 2a. Various other characteristics of a bearing can be studied in a similar manner, an important one from the point of view of practice being that of the thickness of the oil film at the point of nearest approach of the journal to the bearing. This can be expressed as the ratio of the minimum film thickness t to the clearance C, and the dimensional equation written as

$$\frac{t}{C} = \phi_3\left(\frac{ZN}{P}, \frac{L}{D}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

A somewhat similar family of curves is then obtained when t/C is plotted against $\frac{ZN}{P}$ as shown in Figure 2b. It will be noted that the minimum film thickness decreases continuously as $\frac{ZN}{P}$ decreases and this is important in relation to design, since above some limiting film thickness the desirable oil film will remain intact and fluid film lubrication be retained; while below this thickness the film may not be continuous and undesirable wear, excessive friction and possibly seizure may occur. It is useful, however, to get near to this point for normal running conditions, since this means a lower coefficient of friction and therefore less heating and waste of power; the efficient design works just to the right of the minimum point of Figure 2a with a sufficient margin of safety from the minimum point. Since each time the machine is started and stopped the bearing must pass through the left-hand portion and the rather ill defined minimum point, the bearing designer has no easy task.

The brief discussion given above can only be regarded as a simple illustration. No account has been taken of the effect of temperature on the viscosity of the oil or distortion of the bearing under load, and a simple steady loading system has been assumed. Variation of the initial clearance between journal and bearing has not been mentioned; it could be introduced as an additional non-dimensional parameter C/D and studied as such, but this is not necessary since it has been shown that it can be introduced directly into Equations (5) and (6) and there is no need for it to be treated independently. The point it is desired to emphasize is that in the consideration of fluid film lubrication both the experimental method based on dimensional analysis and mathematical treatment of viscous flow can be used, and the former serves to illustrate and check the latter. In addition, by introducing additional non-dimensional parameters representing for example, the effect of temperature changes on oil viscosity, or more complicated types of loading, the experimental method can be applied to cases in which mathematical solutions cannot be obtained.

Fluid Flow in Smooth and Rough Pipes.

A knowledge of the laws of fluid flow in pipes, particularly the pressure loss in long lengths, is of great importance in many branches of engineering practice. As a result of many years of experience much practical data on this subject has been accumulated, but this has usually been in the form of empirical friction coefficients applicable only to particular fluids and pipe surfaces. During recent years, however, the developments of fluid dynamics, both theoretical and experimental, have thrown much light on the real nature of the problem and it can be now be considered on a rational scientific basis. Much still remains to be done in relation to rough pipes, but useful progress is being made, which, in part, at any rate, has practical applications. All engineers are more or less familiar with the difference between laminar and turbulent flow in pipes, and it will be remembered that at low velocities the flow of a liquid through a small straight pipe is characteristically different from that at

higher velocities in larger pipes. At the low velocities the motion is of the viscous or laminar type in which the particles of the fluid move in lines parallel to the axis of the pipe, and lateral transfer of momentum from one line of particles to the next is small and occurs only by molecular action. Above the so-called critical velocity the flow changes to the turbulent type in which there are small rapid irregular velocity fluctuations both along and across the flow direction, and lateral transfer of momentum by the actual movement of small portions of the fluid. It is thus convenient to express turbulent flow in the form of small fluctuating velocities imposed on a mean velocity, both along and across the mean flow direction, and turbulent flow is dealt with mathematically on this basis by statistical methods. Osborne Reynolds in his well-known experiments showed that the change to turbulent flow was dependent on $\frac{vd\rho}{\mu}$ (the Reynolds number for the pipe, d being the pipe diameter and v the mean velocity of the fluid), and occurred at a value of the Reynolds number of about 2000. Actually, the Reynolds number at which the change occurs is not well defined, and it is usual to think of a transition stage between the laminar and turbulent flow regions. Incidentally, a common example of the onset of the change is provided by the smoke rising from a lighted cigarette held in a still atmosphere. The smoke rises for a little distance in smooth lines by laminar flow, and then breaks down into obvious eddies which degenerate into confused turbulent motion.

The viscous or laminar flow of fluids in long pipes is not of major importance to engineers but it does occasionally arise. Thus for water flowing in a 3-inch diameter pipe the critical velocity is only about 0·1 foot per second, but for castor oil in the same size of pipe, the critical velocity would be of the order of 80 feet per second.

Uniform laminar flow in a horizontal pipe provides one case in which the resistance to flow can be determined directly by equating the pressure drop per unit length to the viscous shear forces across the pipe and assuming zero velocity at the pipe wall. The result is usually known as the Hagen-Poiseuille law and written in the form

$$\frac{\delta p}{\delta l} = \frac{32\mu v}{d^2}.$$

The pressure drop per unit length is thus directly proportional to the velocity and viscosity of the fluid. The law can be expressed in the usual engineering form

$$\frac{\delta p}{\delta l} = \frac{\rho v^2}{2d}\left(\frac{64\mu}{\rho v d}\right) = \frac{\rho v^2}{2d}\,\frac{64}{R} \qquad \ldots\ldots\ldots\

velocity fluctuations and the transfer of momentum from layer to layer across the flow, a form of mechanical or apparent viscosity arises which results in an equivalent or apparent shearing stress between adjacent fluid layers very much greater than the viscous stress under similar conditions.

Many years ago Stanton showed that even in turbulent flow in a pipe there is a very thin viscous layer of fluid adjacent to the wall in which there is a steep velocity gradient, the velocity rising rapidly from zero at the wall until it merges into the turbulent motion in the pipe; so that it is only in a very rough pipe that the complete flow can be regarded as turbulent. It has already been pointed out in the discussion of dimensional analysis that the friction or resistance coefficient λ in the ordinary engineering equation for pipe flow is a function of the Reynolds number R in a smooth pipe, and of the Reynolds number and the roughness in a rough pipe. Since R corresponds to the ratio of inertia to viscous forces it follows that in a smooth pipe λ will diminish with increasing R, and in a rough pipe λ will tend to become constant with increasing R and then be dependent only on the relative roughness.

For practical pipe-flow conditions both the thin viscous layer at the wall and the surface roughness must be considered. By a combination of rational ideas on the mechanism of the central turbulent flow and the laminar wall layer with the results of pipe-flow experiments, Prandtl has been able to show that for a smooth pipe the friction coefficient can be expressed in the following form

$$\frac{1}{\sqrt{\lambda}} = 2 \log (R \sqrt{\lambda}) - 0{\cdot}8 \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Von Karman has also obtained a similar expression for smooth pipes; in addition he has shown that when the frictional coefficient is independent of the Reynolds number, the experimental results obtained on pipes artificially roughened by uniform sand grains of various grades are represented by the formula

$$\frac{1}{\sqrt{\lambda}} = 2 \log 7{\cdot}4 \frac{r}{k} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

in which r is the radius of the pipe and k is the height of the sand grains.

The friction coefficient curves for various pipes are shown in Figure 3. The left-hand sloping portion of the curves represents the laminar flow conditions in which roughness has no appreciable effect. At a Reynolds number of about 2000 the change to turbulent flow occurs with an increase in friction coefficient. With a very rough pipe the increase continues at a diminishing rate until the curve is flat and the coefficient is independent of R. A slightly rough pipe follows the smooth pipe law for a considerable distance and then deviates from it until it also eventually becomes independent of R. The degree of roughness determines the Reynolds number at which deviation from the smooth-pipe law commences and with uniform types of roughness such as sand grains, suitable non-dimensional plotting gives coincidence of the curves for various degrees of roughness. The general effect can be visualized qualitatively on the following lines, originally due to Prandtl. It is reasonable, or at any rate plausible, to assume that if the surface irregularities of shape are small and well within the viscous layer they will have no appreciable effect on the resistance to flow, but if they protrude well beyond this layer, the effect of the layer will be negligible and the protruberances will make their own contributions to the general turbulence in the flow.

In determining whether a pipe must be considered smooth or rough for particular flow conditions, the effective thickness of the viscous wall layer is of interest. Necessarily it will diminish with increasing Reynolds

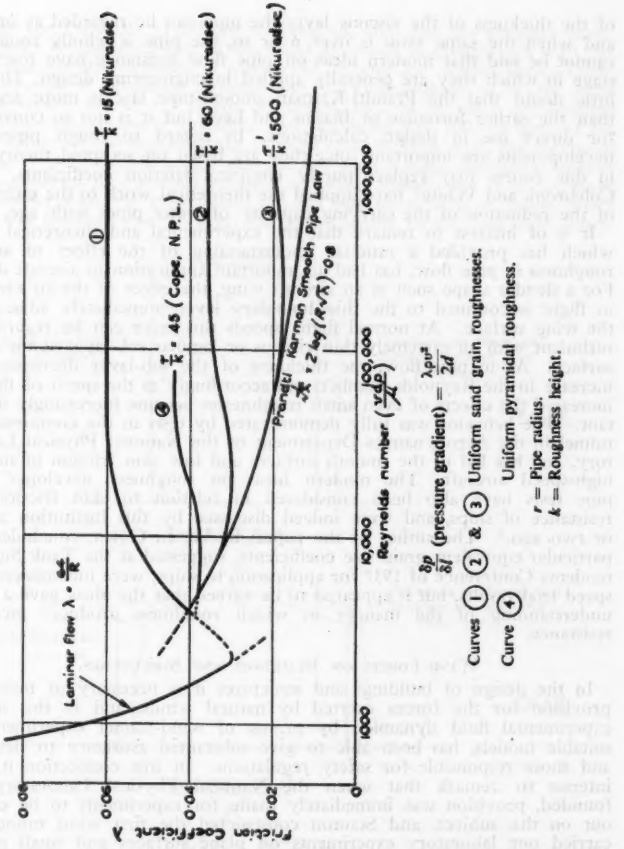

FIGURE 3.—LAMINAR AND TURBULENT FLOW IN SMOOTH AND ROUGH PIPES.

number, and even at a constant value of R should not be regarded as a stable layer since it is continuously shedding eddies and merges over a transition range into the general turbulent stream; however, an approximate idea of the effective thickness is indicated by the following formula

$$\text{Thickness of viscous layer} = \frac{33d}{R\sqrt{\lambda}} \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Thus for water flowing at a mean velocity of 10 feet per second in a 12-inch pipe the thickness is about 0·004 inch. It has been suggested that when the height of the roughness particles is less than about ¼

of the thickness of the viscous layer the pipe can be regarded as smooth, and when the same ratio is over 6 or so, the pipe is wholly rough. It cannot be said that modern ideas on pipe flow resistance have reached a stage in which they are generally applied in engineering design. There is little doubt that the Prandtl-Karman smooth-pipe law is more accurate than the earlier formulae of Blasius and Lees, but it is not so convenient for direct use in design calculations. In regard to rough pipes, the developments are important since they are based on accepted theory, and in due course may replace purely empirical friction coefficients. Thus Colebrook and White[2] have applied the theoretical work to the estimation of the reduction of the carrying capacity of water pipes with age.

It is of interest to remark that the experimental and theoretical work which has provided a rational understanding of the effect of surface roughness in pipe flow, has had an important application in aircraft design. For a slender shape such as an aircraft wing, the effect of the air viscosity in flight is confined to the thin boundary layer immediately adjacent to the wing surface. At normal flight speeds this layer can be regarded as turbulent with an extremely thin viscous or laminar sub-layer at the actual surface. As in pipe flow, the thickness of the sub-layer decreases with increase in the Reynolds number, and accordingly as the speed of flight is increased the effects of even small roughnesses become increasingly important. The behavior was fully demonstrated by tests in the compressed-air tunnel in the Aerodynamics Department of the National Physical Laboratory, and has led to the smooth surfaces and low skin friction of modern high-speed aircraft. The modern ideas on roughness developed from pipe tests have also been considered in relation to skin friction and resistance of ships, and were indeed discussed by this Institution a year or two ago.[3] The author of the paper, R. W. L. Gawn, concluded that particular equivalent grain size coefficients, suggested at the Tank Superintendents Conference of 1937 for application to ships, were inconsistent with speed trial results, but it appeared to be agreed that the ideas gave a fuller understanding of the manner in which roughness produces increased resistance.

Wind Forces on Buildings and Structures.

In the design of buildings and structures it is necessary to make due provision for the forces exerted by natural winds, and in this matter, experimental fluid dynamics, by means of wind-tunnel experiments on suitable models, has been able to give substantial assistance to designers and those responsible for safety regulations. In this connection it is of interest to remark that when the National Physical Laboratory was founded, provision was immediately made for experiments to be carried out on this subject, and Stanton constructed the first wind tunnel and carried out laboratory experiments on plane surfaces and small models of buildings and girders in a uniform current of air. The results obtained were communicated to the Institution of Civil Engineers just forty years ago[4], and were welcomed by the President (Sir William White, the eminent naval architect) as the first contribution from the National Physical Laboratory recording researches made there which had a bearing on engineering work. There is little doubt that these particular experiments, primarily made with the object of obtaining information on a subject of immediate practical interest to civil engineers, formed the foundation of experimental aerodynamics in this country.

In estimating the action of the wind on an exposed building it is necessary to consider both the extent and distribution of the wind-pressure forces and also the summation of these expressed as one or more resultant forces acting on the complete structure. In making use of model experiments for the determination of such forces it is at once evident that the

flow in the full-scale prototype is turbulent, and that the pressure variations over the building due to the wind are small in comparison with the mean atmospheric pressure of about 14·7 pounds per square inch. Complete dynamical similarity requires the Reynolds number to be the same in the model as the full scale prototype, but, as in most wind-tunnel experiments, this is not practicable. For a building, say, 100 feet deep or for a gas holder of 100 feet diameter exposed to a 50 Mph. gale, the Reynolds number is $4{\cdot}5\times10^{7}$; it is not possible for such a high Reynolds number to be obtained in a normal atmospheric wind tunnel, and it is necessary to consider how far experiments carried out on models at much lower Reynolds numbers can be regarded as applicable to the full-scale condi

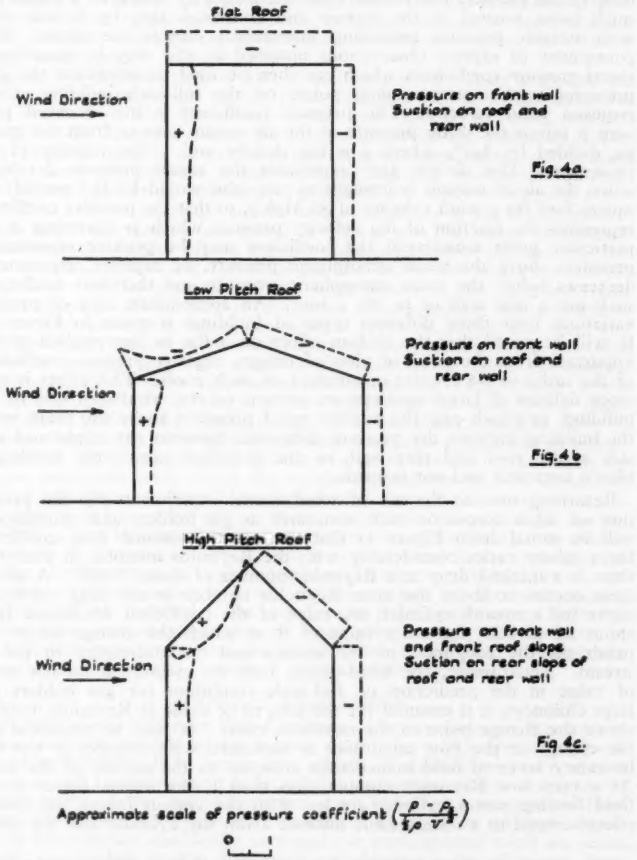

FIGURE 4.—WIND PRESSURE ON BUILDINGS.

tions. Fortunately, experiment shows that in the case of bluff bodies such as square or rectangular buildings, the force coefficients and the pressure distribution, while dependent on the shape of the building and the direction in which the wind is blowing, are largely independent of the Reynolds number. Accordingly, for such buildings, wind-tunnel tests can be carried out on fairly small models, say a few inches high, and the results applied to the full-scale conditions with a fair measure of confidence. The conditions are not so simple, however, for cylindrical structures such as chimney stacks and gas holders, and reference will be made to this later.

In wind-tunnel tests on model buildings, the usual procedure is to observe the pressure distribution over the model by means of a number of small holes normal to the surface and communicating by flexible tubes with suitable pressure measuring instruments outside the tunnel. It is convenient to express observations obtained in this way by non-dimensional pressure coefficients which can then be used to determine the local pressures at the corresponding points on the full-scale building at any required wind velocity. The pressure coefficient is the observed pressure p minus the static pressure in the air stream remote from the model ps, divided by $\frac{1}{2}\rho v^2$, where ρ is the density and v the velocity of the air-stream. The divisor $\frac{1}{2}\rho v^2$ represents the excess pressure developed when the air in motion is brought to rest (this would be 16·4 pounds per square foot for a wind velocity of 80 Mph.), so that the pressure coefficient represents the fraction of the velocity pressure which is operating at the particular point concerned; the coefficient may be positive representing pressures above the mean atmospheric pressure, or negative, representing decreases below the mean atmospheric pressure and therefore tending to suck out a rear wall or to lift a roof. An approximate idea of pressure variations over three different types of buildings is given in Figure 4.* It will be noted that the suction effect on a flat or low-pitched roof is important from the point of view of design; negative pressure coefficients of the order of 0·8 are not uncommon on such roofs. This effect is even more definite if large openings are present on the windward side of the building, in which case the positive wind pressures above the mean inside the building increase the pressure difference between the inside and outside at the roof and rear wall, so that if failure occurs the building is blown outwards and not inwards.

Returning now to the use of wind tunnel experiments for the prediction of wind forces on such structures as gas holders and chimneys, it will be noted from Figure 1a that the non-dimensional drag coefficient for a sphere varies considerably with the Reynolds number; in particular there is a marked drop at a Reynolds number of about 3×10^5. A similar drop occurs at about the same Reynolds number in the drag coefficient curve for a smooth cylinder, the value of the coefficient decreasing from about 1·2 to 0·35; the exact value of R at which the change occurs depends on the roughness of the surface and the turbulence in the air stream. Accordingly, for wind-tunnel tests on cylindrical models to be of value in the prediction of full-scale conditions for gas holders and large chimneys, it is essential for the tests to be made at Reynolds numbers above the change point in the resistance curve. It may be remarked that the change in the flow conditions is associated with changes in the thin boundary layer of fluid immediately adjacent to the surface of the body. At a very low Reynolds number, less than 1, the inertia forces in the fluid flowing past a cylinder are less than the viscous forces, the viscous effects extend to a considerable distance from the cylinder and the resist-

* The information given in Figure 4 is partly derived from experiments by Bailey and Vincent, Journ. Inst. Civil Engrs., Oct., 1943.

ance is approximately proportional to the speed. The flow phenomena are then similar to the flow of a very viscous fluid in a pipe or oil in a fully lubricated bearing. At much higher Reynolds numbers but below the change point, the effect of viscosity is confined to the thin viscous or laminar boundary layer which separates from the surface towards the rear of the cylinder in the form of eddies and vortices which are dissipated in the wake. Finally, at Reynolds numbers above the change point the boundary layer itself changes from laminar to turbulent flow before it reaches the breakaway point on the cylinder and thus causes the breakaway to move around towards the rear, decreasing the wake and the resistance coefficient. In general the change from the laminar to turbulent boundary layer is akin to the change from laminar to turbulent flow in a pipe, but in the latter case the turbulent boundary layer extends across the full cross-section of the pipe at distances sufficiently remote from the entrance.

Figure 5 is a photograph of a 1/40 scale model of a spirally guided gas holder recently made and tested at the National Physical Laboratory for the Institution of Gas Engineers; in this case tests at Reynolds numbers as high as 2×10^6, well above the change point in the resistance curve, were possible, and both pressure distribution experiments and balance measurements of resultant forces were made. The distribution of pressure around cylindrical structures exposed to the wind is frequently important in relation to design and an indication of the types of distribution which may be expected are given in Figure 6. The exact distribution varies with the height of the section considered above the ground, but for a smooth surface and Reynolds number below the change point the distribution will be generally of the form shown in the right-hand half of the diagram. Over part of the front of the section the pressure is positive and above the mean atmospheric pressure, but at the sides and rear there are marked suction effects, particularly at the sides. At high Reynolds number and with rough surfaces or where there are external frameworks the pressure distribution changes to the form shown in the left-hand portion of the diagram, the suction at the sides being much less marked than in the previous case. It seems likely that this form of pressure distribution is reasonably correct for large cylindrical structures exposed to high winds, although there is little detailed information on full-size structures with which the model experiments can be compared.

Wind resistance of ships is of considerable importance to naval architects and model experiments on this matter have been carried out by Hughes[6] of the William Froude Laboratory. It is not possible to discuss the results in the present lecture, but it is of general interest to mention that the experiments were made on ship superstructure models, towed, upside down, through water at speeds sufficiently low for the wave-making resistance to be neglectd. The tests were thus made on a Reynolds number basis, and they provide an instructive example of the way in which model experiments can be carried out in one fluid although the full-scale conditions are concerned with a different fluid.

River and Tidal Models.

It is with some diffidence that I include a few remarks on the subject of river and tidal models in this lecture, since I have little personal experience of such work. It is, however, a subject of much importance to civil engineers, and for this reason is included in a lecture which endeavors to illustrate some of the more important ways in which experimental fluid dynamics is applied to engineering practice. There is no national laboratory in this country specifically arranged for experimental work on river, estuary, and harbor models, but much valuable work has of course been carried out here, particularly at the University of Manchester, first by

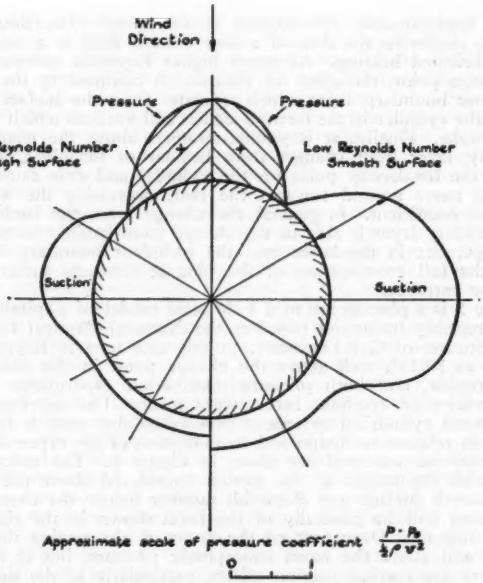

FIGURE 6.—WIND PRESSURE ON CYLINDRICAL STRUCTURES.

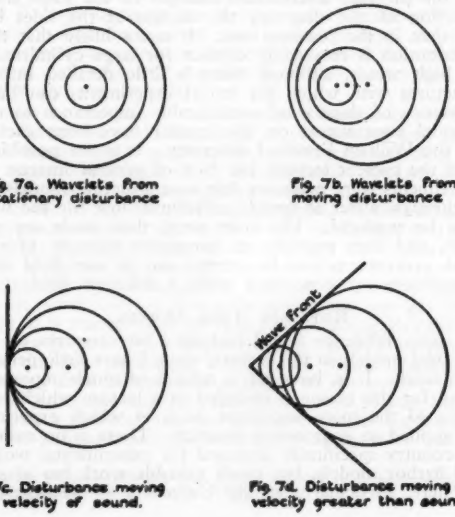
Fig. 7a. Wavelets from stationary disturbance
Fig. 7b. Wavelets from moving disturbance
Fig. 7c. Disturbance moving at velocity of sound.
Wave Front
Fig. 7d. Disturbance moving at velocity greater than sound.

FIGURE 7.—SMALL DISTURBANCE MOVING IN STILL AIR ILLUSTRATING FORMATION OF SHOCK WAVE.

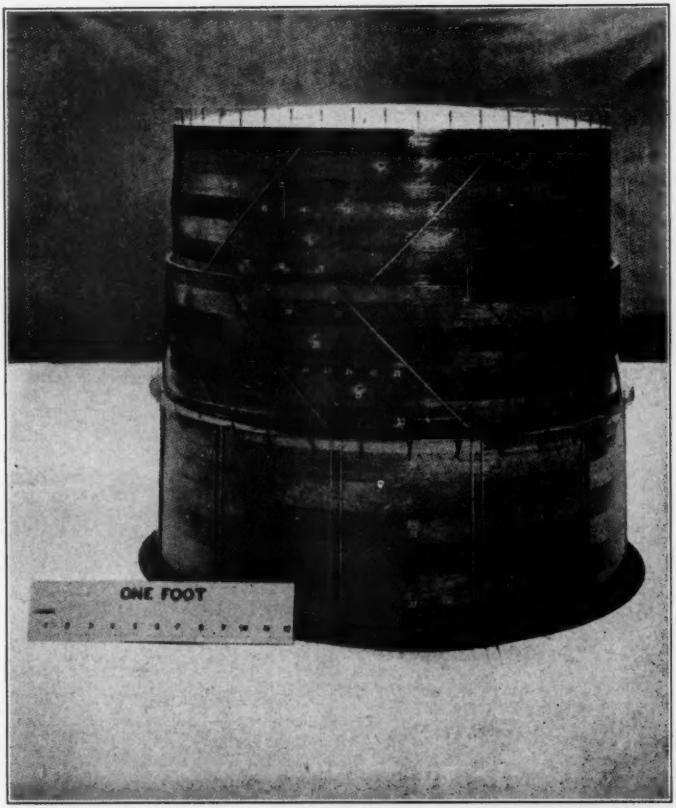

FIGURE 5.—1/40 SCALE MODEL GAS HOLDER USED IN WIND TUNNEL TESTS.

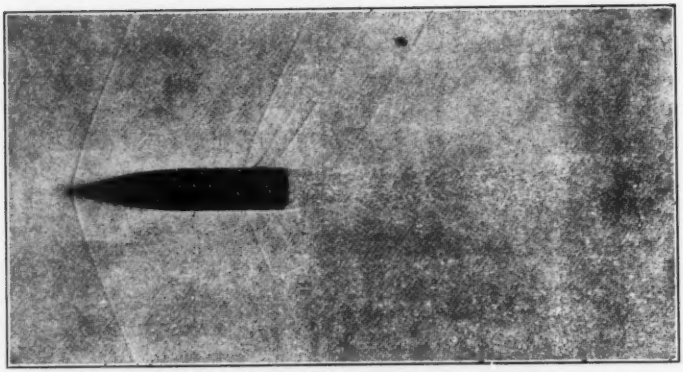

FIGURE 8.—PROJECTILE IN FLIGHT SHOWING SHOCK WAVES.

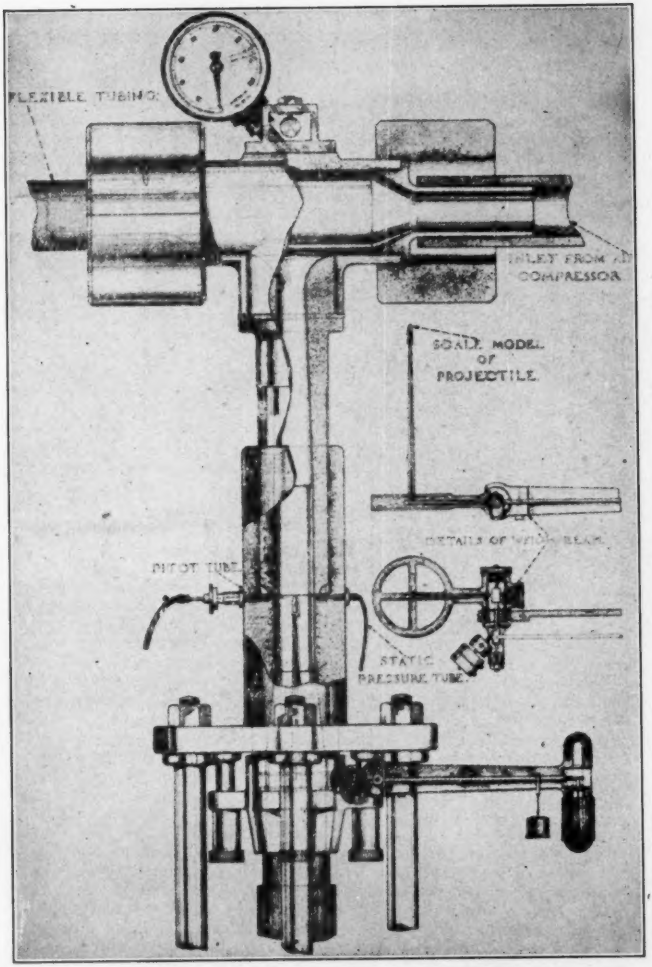

FIGURE 9.—STANTON'S ORIGINAL SUPERSONIC WIND TUNNEL.

Osborne Reynolds, and subsequently by Professor A. H. Gibson. Indeed, the work carried out by the latter for the Severn Barrage Committee[6] provides an outstanding example of the manner in which model experiments on tidal flow can be used in the study of a proposed engineering project of great magnitude and national importance. In some other countries where problems of river flow are perhaps more important than in Great Britain, national laboratories for the study of such problems have been in existence for many years, and a description of the principal European laboratories is given in "Hydraulic Laboratory Practice" published in English in 1929 by the American Society of Mechanical Engineers. Since that date the U. S. Government has established a Waterways Experiment Station near the Mississippi, and there have been parallel developments in other parts of the world.

In water flow between solid boundaries and with a free surface, the controlling forces are inertia, gravity and viscosity,* and accordingly complete dynamical similarity requires the same Froude and Reynolds numbers in the model and full-scale prototype in addition to geometrical similarity. This is not practicable when water is used in both systems, and many water-flow models are operated on a constant Froude number basis, care being taken that the velocities in the model system are sufficiently high to produce fully turbulent flow. This method is applied to the experimental solution of such problems as the flow of water over weirs, through openings and sluices, and to those in which a standing wave is produced. The corresponding time scale between model and full scale is then $\sqrt{r}$ where r is the linear ratio of the model to the full-scale prototype, and forces, discharges, energy, etc., in the prototype can be obtained from the model experiments.

The chief use of models of river estuaries lies in the study of the variations produced by proposed engineering works on the general flow and the resulting changes in shoal and silt formations in the various channels and waterways. Accordingly, the problems encountered are concerned not only with water flow, but also erosion of banks, and the transport of sediment and bed materials. The use of such models offers the great attraction that the experimenter is not only able to examine the immediate effect of proposed changes, but can forecast the effect of the changes many years hence. In the construction of the models it is not practicable to use the same scale for both longitudinal and vertical distances, and in the Severn tidal flow model Gibson used a horizontal scale of 1/8500 and a vertical scale of 1/200. This necessarily means a departure from dynamical similarity but does not appear to invalidate the practical usefulness of the method. The time scale for the tidal ebb and flow of estuary models is based on the correct reproduction of the tidal wave, and since the wave velocity is proportional to the square root of the depth of the water in which it is travelling, the tidal period is fixed by the horizontal and vertical scales which have been adopted.† It has been found by experiment that results in accordance with nature are reproduced in tidal models when the bed material is about the same size or slightly less than in the full-scale estuary, and this, in conjunction with the convenient time scale, enables the effect of many years of natural tides to be reproduced in the model in a comparatively short time. Thus in the Severn model the tidal period was 74 seconds, and Gibson was able to start with a model bed corresponding to a survey made in 1849,

* Surface tension cannot always be neglected.

† $\frac{t}{T} = \frac{l}{L}\sqrt{\frac{H}{h}}$ where $\frac{t}{T}$ are corresponding times, $\frac{l}{L}$ the horizonal scale ratio, and $\frac{h}{H}$ the vertical scale ratio of model and prototype.

and by applying the correct number of tides, to show that his model correctly reproduced the natural conditions found in a subsequent survey in 1927. From the exact scientific point of view it is clear that the problem of transport of bed materials in a moving stream is very complex; necessarily it is concerned with the turbulence, velocity gradient and tractive forces near the fluid boundary, with the shape, size and weight of the solid particles, as well as the slope of the bed and percolation of the fluid into the bed material. Rational ideas on this difficult problem are being developed; in the meantime, the successful manner in which tidal models have been shown to reproduce natural conditions fully demonstrates their value in civil-engineering practice, despite the departure from dynamical similarity inherent in the differing horizontal and vertical scales.

THE EFFECT OF ELASTICITY OR COMPRESSIBILITY IN HIGH SPEED FLUID MOTION.

The obvious problem in fluid dynamics in which compressibility is highly important is that of the resistance to motion experienced by a projectile. The fundamental difference between resistance at low speeds and at supersonic speeds has long been realized; in the former case the energy is largely lost in eddies in the wake, whereas in the latter it can be dissipated in the waves which accompany the projectile in its flight. Again, in aerodynamic research in recent years much attention has been given to problems of motion through the air at high speeds because of the need of increased knowledge of the behavior of high-speed propellers, and in view of the possibility of the speed of the aeroplane itself reaching such high values that changes in resistance and lift due to compressibility must be considered. In general, the modern developments of these branches of fluid dynamics, sometimes known as gas dynamics, are very much the domain of specialists, and from their military nature are not well known to the average engineer. It is, however, my personal view, that the knowledge of the behavior of gases in motion at high speeds, derived from ballistics and aerodynamics, is reaching the stage at which it is likely to be of real assistance to mechanical engineers, particularly those concerned with the design of high-speed steam and gas turbines, and air compressors; for this reason I think an outline of the subject should form a not unimportant part of the present lecture. Moreover, in these days when most of us have a personal knowledge of bombs and shells as distinct from our professional activities, the subject is not without some general interest.

If we think of a small disturbance in still air the wavelets generated can be imagined as proceeding outwards in concentric spheres moving with the velocity of sound as shown in Figure 7a. If the source of the disturbance is moving in a straight line at a low velocity, the wavelets will move outwards from the source as shown in Figure 7b, and if the disturbance itself be travelling at the velocity of sound the wavelets can be imagined as in Figure 7c. Finally if the speed of the disturbance is greater than the velocity of sound (about 1100 feet per second), the wavelets will tend to be left behind and a positive wave front will be built up as a cone proceeding from the disturbance, Figure 7d; obviously the angle of the conical wave will depend on the speed of the disturbance. Waves of this type, known as shock waves, accompany a projectile when it is moving through the air at supersonic velocity, and can be photographed; an example is given in Figure 8. A shock wave must have a finite thickness, but this is very small, and it can be conveniently regarded as a discontinuity in which the change is not adiabatic; thus there is a sudden increase in pressure and temperature in passing from the front to the back of the wave, sufficient for the density change to show on a photographic plate.

It is at once apparent that supersonic wind tunnels are likely to be of value in experimental ballistics, but it was not until after the last war that a real attempt was made to tackle this difficult problem. In 1922 the first supersonic wind tunnel was made in the Engineering Department of the National Physical Laboratory, the ideas embodied being almost entirely due to the late Sir Thomas Stanton. A reproduction of his descriptive diagram of this tunnel is shown in Figure 9. It will be recalled by those familiar with steam nozzles that if a gas be allowed to expand adiabatically from a high pressure through a converging-diverging nozzle the mass flow is controlled by the cross-sectional area of the throat, and the pressure, velocity and temperature at any section of the nozzle can be calculated from the well known adiabatic relationships and the area at the particular section considered. Thus in the converging part of the nozzle the velocity increases rapidly until it reaches the velocity of sound at the throat, the pressure then being of the order of one-half the original pressure. In the diverging portion, the velocity, now supersonic, continues to increase and the pressure and temperature to diminish. If, after a certain amount of divergence, the nozzle area be made constant, uniform flow at a constant supersonic velocity should be obtained. This is the basic principle on which supersonic wind tunnels are designed; obviously the velocity attained will depend on the extent to which the divergence or expansion has been continued beyond the throat, and accordingly for each required speed a separate nozzle is usually necessary. The motive air for Stanton's original tunnel was derived from a 50 Hp. air compressor, delivering 200 cubic feet of free air per minute at a maximum pressure of 120 pounds per square inch, and his own words can well be quoted[7]:— "After a series of preliminary tests, it was found that by allowing the compressed air from the receiver to expand through a converging-diverging nozzle, terminating in a parallel part, to a pressure about one-twelfth its original value, it was possible to maintain continuously in the parallel part which was 0·8 inch diameter, an air current moving at above twice the speed of sound." It will be noted from the diagram that for this tunnel, only 0·8 inch diameter, a balance was constructed on which a model projectile could be mounted; these projectiles were 0·09 inch diameter, and although the resistance measurements were not very accurate, the complete tunnel and balance served to demonstrate that a continuously operated supersonic wind tunnel was practicable. Subsequently Stanton designed a 3 inches diameter supersonic wind tunnel operated from a much larger compressor delivering air at 80 pounds per square inch, and carried out a number of tests on projectiles and aerofoils at speeds up to 3¼ times the velocity of sound. In Stanton's tunnels the motive air passed through the tunnel from a relatively high pressure and exhausted to the atmosphere; in more recent supersonic wind tunnels such as those constructed in Switzerland and Italy[8], the air circuit has been enclosed, the exhaust air returning to the compressor, and working sections up to 16 inches square have been used; the horsepower of the Italian tunnel has been stated to be about 3000.

The existing supersonic wind tunnels have been built primarily for experimental work in ballistics and aerodynamics, and in the actual tunnel the diverging portion of the nozzle is not a straight taper but is specially shaped in order to obtain uniform air-flow conditions along and across the working section in which the model is placed. In developing the correct shape of these nozzles considerable use is made of photographic methods for the detection of shock waves, and of search tubes of the static and total head type. Accordingly, it is of interest for the mechanical engineer to realize that detailed information is being accumulated on the flow of air in large convergent-divergent nozzles, in the Italian tunnel possibly several feet long and up to 16 inches across, and it seems this

information must in due course prove to be of value in the design of turbine nozzles. Again, it has been pointed out by Guy[9] that although air-flow experiments through nozzles are of assistance to turbine designers, they are usually carried out at speeds at which the effect of compressibility is small, whereas under actual steam-flow conditions the effect of compressibility may be very marked. A supersonic wind tunnel appears to provide a convenient apparatus in which nozzle experiments could be carried out in which not only the Reynolds number, but also the Mach number could be related to actual turbine conditions. It may be mentioned that for true similarity between the flow of two gases in which compressibility effects occur the value of γ (the ratio of the specific heats) should be the same for both. Thus it would not be fully correct to apply the results obtained in air-flow experiments to steam-flow practice. If, however, the experiments were made with say, carbon dioxide, a matter which should not be impossible in a closed-circuit return-flow supersonic wind tunnel, the results should be directly applicable to steam flow.

Another point of interest to the mechanical engineer which arises from the operation of a supersonic wind tunnel is that of diffuser efficiency. The high velocities are obtained by the adiabatic expansion of the air through a convergent-divergent nozzle which transforms the initial intrinsic energy into velocity in the working section. Unless a reasonable proportion of the velocity energy can be reconverted into pressure in a suitable diffuser, excessively high pressures are required from the air compressor driving the plant. Much information is available concerning sub-sonic diffusers and efficiencies of the order of 90 per cent can be obtained by tube or duct walls which diverge at an angle of about 8 per cent; it should be appreciated, however, that theoretically, in a supersonic diffuser, the walls should first converge to a throat to reduce the velocity to the sonic velocity and then diverge in the ordinary way for a sub-sonic diffuser. A further point of interest is that in supersonic flow a shock wave can be used as a diffuser. It has already been stated that in passing from the front to the back of the shock wave accompanying a projectile in flight there is an increase in pressure and temperature. If now a gas is flowing in a tube or channel at supersonic velocity and a shock wave occurs, there will be an immediate increase in pressure and decrease in velocity at the shock, and it has acted as a somewhat inefficient diffuser.

In regard to the use of supersonic wind tunnels in ballistic research, little will be said at the present juncture. A typical resistance coefficient curve against Mach number for a projectile has already been shown in Figure 1c, and it will have been noted that the value rises from about 0·2 at sub-sonic speeds, up to about 0·6 just above the velocity of sound, and then diminishes. The energy lost in the head and other waves accompanying the projectile largely accounts for the increased resistance at the high speeds. It may be mentioned, however, that the resistance coefficient at the high speeds, while very largely a function of the Mach number, may also be dependent to a small extent on the Reynolds number; in this, and in the wave-making resistance, the members of this Institution will no doubt recognize the similarity (using the word in its general sense) between supersonic wind-tunnel and ship tank-model experiments. It should be realized that both the pressure and temperature in the working section of a supersonic wind-tunnel are much below atmospheric; thus in Stanton's 3-inch tunnel when running at a Mach number of 3, the pressure and temperature in the working section are about 2 pounds per square inch absolute and −160 degrees C. respectively. This big difference from atmospheric conditions will have no effect on the applicability of the results to the full scale as far as elasticity or compressibility effects are concerned, but may have some small effect in regard to viscosity effects; thus the Reynolds number for the model in the 3 inch tunnel

(based on projectile length) is about one million, whereas the full-scale Reynolds number for a 12-inch diameter shell may be 100 times as great. From the statement by Stanton[7], however, that fair agreement has been obtained between model tests and firing trial results, it may be concluded that the Reynolds number effect is small.

The general shape of the resistance coefficient curve for a projectile, rising rapidly to a maximum just beyond the velocity of sound, obviously applies to bombs dropped from aircraft. An elementary calculation, based on, say, the maximum resistance coefficient of the projectile of Figure 1c suggests that a fairly heavy bomb is required for the velocity, when the bomb is approaching the earth, to exceed the velocity of sound; the interesting point to us as civilians at the present time is that unless the velocity exceeds the velocity of sound, the bomb will be heard approaching.

It will no doubt have been appreciated from the foregoing remarks on ballistics, that at speeds which approach the velocity of sound the overall resistance of an aircraft and the characteristics of an aerofoil are likely to undergo a profound change. This is indeed the case, and as an example, experiments by Stanton[10] show that with the aerofoil known as RAF31a, the maximum value of lift/drag diminishes from 20 at a Mach number of 0·5 (corresponding to about 370 miles per hour), to 10 at a Mach number of 0·7 (520 miles per hour), while above the velocity of sound this aerofoil would be woefully ineffective. The marked increase in drag and the collapse of the lift coefficient at high speeds is obviously a fundamental obstacle to very great increases in aeroplane speed, at any rate, on orthodox lines. In recent years mechanical engineers have become interested in modern aerofoil theory and experiment in view of its practical possibilities in turbine- and air-compressor blade design. It is clear, however, from the marked changes which compressibility produces in aerofoil characteristics, that when high speeds occur consideration of blade behavior on the Reynolds number criterion alone may not be sufficient, and that model blade tests may be far from correct unless they have been carried out at Mach numbers corresponding to the prototype design. The study of aerofoil and propeller performance at high speeds is being actively pursued in aerodynamical laboratories, and for this purpose high-speed wind tunnels have been developed which are capable of air speeds of several hundred miles an hour, although they are not designed to operate as supersonic wind tunnels. The position reached in 1938 was reviewed by Lock[11] who also gave details of the high-speed wind tunnel operated from the exhaust of the compressed-air tunnel in the Aerodynamics Department of the National Physical Laboratory.

In concluding this necessarily brief survey of the effect of compressibility, it is of interest to remark that the fundamental equations of fluid dynamics which involve compressibility have been shown to be less difficult to solve than the equations involving viscosity. Thus Taylor and Maccoll[12] have been able to calculate pressure distribution on cones moving through a gas at supersonic speeds, and draw interesting conclusions, verified by experiment, on the position and angles of the accompanying shock waves; approximate methods of calculation of lift on thin aerofoils at supersonic speeds have also been developed.

Concluding Remarks.

In presenting this lecture to you my main object has been to state certain fundamental principles on which experimental fluid dynamics is based, and then to illustrate these principles by examples of their application to engineering practice other than aeronautics and naval architecture. Largely owing to limitation of time, I have omitted all reference to a branch of fluid dynamics of great importance to mechanical engineers, namely, heat transmission; I would remark, however, that an

admirable summary of this subject was recently presented by Lander[18] to the Institution of Mechanical Engineers. Necessarily, the number of applications to engineering practice which I have been able to discuss has been small and as a final illustration I have prepared a table, which although it is far from exhaustive, gives a general idea of various branches of engineering in which fluid flow experiments carried out on models have been, or are likely to be, of value in practice. It should be noted that the model is not, of necessity, smaller than the full-scale prototype; it may be of the same size, or in some cases, such as turbine blades, larger than the full scale. Neither is it essential for the model to include all details of the prototype, or for the same constructional materials to be used; frequently, it may be made of cheaper material, in which small alterations of shape, etc., can conveniently be made, and their effects examined; and it is sometimes convenient to use a different fluid in the model experiments. The essential requirement is correct application of the laws of dynamical similarity and dimensional analysis, and when deviations from these are unavoidable, and this is not infrequently the case, for reliable check tests to be made by full-scale experiments.

PRACTICAL APPLICATIONS OF EXPERIMENTAL FLUID DYNAMICS.

Application	Controlling Parameter Reynolds number = R Froude number = F Mach number = M
Lubrication	Viscous flow
Flow of molten metal in furnaces, etc.	Viscous flow
Penetration of water through soils, filters, etc.	Viscous flow
Aerodynamic forces on aircraft and aircraft components	R (R and M for high speeds)
Aircraft propellers	R (R and M for high speeds)
Fans	R
Wind pressure on buildings, chimneys, gas holders, etc.	R
Air resistance of motor cars, trains, ships	R
Ventilation of buildings, mines, etc.	R
Water propellers	R
Water turbines	R
Centrifugal pumps	R
Valves for all fluids	R (sometimes M)
Flow of all fluids in pipes and closed ducts	R
Resistance of submarines, torpedoes, etc.	R
Hydraulic recuperators	R
Fluid shock absorbers	R
Drowned orifices	R
Flow meters	R (M occasionally)
Ship launching	F
Weirs and spillways, open orifices	F
River and canal flow	F and R
Open ducts for water	F and R
Resistance of ships	F and R
Gravity mixing of fluids	F and R
Aerodynamic forces on projectiles	M
Steam and gas turbine blades	M and R
Steam and gas turbine nozzles	M and R
Steam injectors	M and R
Steam and gas flow in engine ports, valves, etc.	M and R

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A PLEA FOR THE REVISION OF THE TONNAGE LAWS.—This is the Presidential Address of A. Murray Stephen, delivered before a meeting of the Institution of Engineers and Shipbuilders in Scotland, on October 12, 1943. It is reprinted from the Transactions of that Institution for November, 1943. Mr. Stephen deals with a very complex problem in such a simple and straightforward manner as to make the subject understandable to those not possessing knowledge of the involved subject.

It has been usual for the President at the opening of a new session to give an address on a subject of which he has a special knowledge. But I feel that on this occasion I ought to devote my attention to a problem which has, for forty years at least, been discussed from time to time by this Institution and other similar bodies, and which appears to be even further than ever from a satisfactory solution. I refer to rules for the measurement of tonnage of ships. A unique opportunity appears now to be approaching to achieve a revision of these rules. If this chance is missed the probability is that never again will a set of circumstances arise so favorable for a general revision.

After the war by far the largest merchant fleets will be possessed by the United States of America and by the British Commonwealth of Nations. Moreover, of the two great canals used by the ships of all nations, the Panama is controlled by the United States and the Suez by the French and British. In addition, most of the chief maritime nations of the world are working with us in this war. An overwhelming majority of the nations that are most vitally concerned with this problem are, therefore, already linked together in a common cause against Germany and Japan. From the point of view of the shipowner, too, this would appear to be a unique opportunity for a change, if a change is ever to be made. For, almost universally, their fleets are woefully diminished and they are faced with the necessity of rebuilding them over the next few years if they are to maintain their trade. Consequently a change now would have far fewer repercussions than in normal times. But even more important is

the psychological attitude abroad among the nations today. When everything is in the melting pot men are prepared to examine and consider any change. But when we all slip back again into the smooth stream of peace men will be less inclined to look with favor on anything which may cause even the slightest ripple of inconvenience on the surface.

The tonnage laws provide for two measurements of tonnage: the gross tonnage—roughly a measurement of the cubic capacity of the ship and her superstructure—and net tonnage, which is supposed to represent the cubic capacity of the spaces in the ship which earn money. For the purposes of assessing the ship for dues the net tonnage is usually used, though for some purposes, such as dry dock dues, the gross tonnage is sometimes applied. It would therefore appear that, in general, the port dues are assessed on what was supposed to be a measure of the earning capacity of the ship. But is this the right principle to apply? In the rendering of most other services the payments are not based on this principle, but are based roughly on the cost to the supplier of rendering that particular service. So there are strong grounds for challenging the very principle upon which the present tonnage laws are based.

I will not attempt even to sketch the evolution of the tonnage laws, as this has been dealt with many times before. Suffice it to say that, while the early laws were framed in an endeavor to provide a fair basis for taxation as between ship and ship, these early laws have been so amended and overlaid with compromise that in the discussions on the subject no one has ventured to defend them on the ground that they are fair. But apart from their fairness or otherwise, they are now very complicated and require the measurement of most spaces in the ship. It would be interesting to know how many surveyors all over the world spend the greater part of their lives measuring ships for tonnage, an occupation in which there might be some satisfaction if it were the only way to achieve fairness, but of which under the present rules this cannot be said. If a simpler and fairer set of laws can be found—and I am sure that it can—then those highly skilled men are simply wasting their energies on the altar of an outworn conservatism, and not only their energies but the shipowners' and the nation's money.

Because the present tonnage laws are unfair as between ship and ship, shipowners and builders are forced to produce ships which are in many respects neither as efficient nor as seaworthy as they might be. Consequently, if a ship does not take advantage of some of the loopholes in the law, she is penalized and has to pay higher dues all her life. The most important of these so-called loopholes, and the one most likely to have important effects on the future, is the way in which machinery spaces affect tonnage. If the machinery spaces of a ship, including light and air trunks, and shaft tunnels, amount to 13 per cent or more, and under 20 per cent of the gross tonnage of the ship, then a 32 per cent deduction is obtained from the gross tonnage in arriving at the net tonnage. Now, modern machinery is becoming lighter and lighter and more and more compact and we have already reached the stage where the designer has often, in order to get up to the 13 per cent, to provide large spaces in the center of the ship which are simply wasted. He has even sometimes to build up his engine or boiler hatch above the boat deck, sticking up into the air, in order to increase the entire space which comes into machinery space measurement. As machinery becomes smaller and still more compact, as it undoubtedly will, many a ship will carry round with her all her life a large useless, almost empty space in her middle. But it is hidden and no one sees it, and those who sail in ships have become so used to it that they never think about it. If these hidden empty spaces had been visible, like a funnel or a mast, public opinion would, before now, have forced a revision of the rules. A distinct brake is thereby put

on the progress of marine engineering in that a part of the benefits which smaller and lighter machinery should reap is withheld. Admittedly, some limiting factor was necessary in the past, and may be necessary in the future, to ensure that working spaces such as engine and boiler rooms are ample for the health and well-being of the men who work in them, but surely this is a function which the Board of Trade should exercise quite apart from tonnage measurement.

Another entirely illogical situation has been created by the effect of the measurement of superstructures for tonnage. Originally, no doubt with the laudable idea of encouraging as much enclosed deck shelter as possible, superstructures which were not totally enclosed were exempted from all tonnage measurement. But as time went on the superstructure became larger and the apertures to them smaller until the position became quite topsy turvy; it now amounts to this, that if one has a complete superstructure one may, by making apertures in it, and by fitting scuppers to it, thereby making the ship less seaworthy, render it exempt from net tonnage. It is true that these apertures may be boarded up, but the regulations strictly forbid any sort of door or permanent form of fastening, if the exemption from net tonnage is to be retained. The details of the application of this regulation as to when such a superstructure may or may not be exempt reach such a fine point of delineation as to be ridiculous. In the case of a river steamer, I am told that the decision for or against the inclusion of a deck shelter in tonnage almost turned on whether curtains and aspidistras were supplied or not, and I can well believe it.

The commonest type of ship affected in this way is that known as the shelter deck ship. In this type the shelter deck superstructure covers the whole deck of the ship and the shelter deck has in effect become the weather deck of the ship. But, if a tonnage opening is left in the deck just large enough to fulfill the definition of such an opening, the whole of this shelter deck space is exempt from net tonnage except such space as happens to be occupied by cargo. The tonnage laws have here offered a very heavy bribe to owners and designers and a large number of shelter deck ships have been built. So far has this idea been developed that in some recent coasters the shelter deck space looks as though it actually were more than the remainder of the cargo space. During the war these shelter deck ships have their tonnage openings closed by means of a permanent and sound type of structure and have been enabled to take advantage of increased draught in consequence. It is possible that owners of shelter deck ships would be averse to any change in the tonnage laws which had given them these advantages, and that there are so many of these owners that they would be able to defeat any change. But is this not a short-sighted view? Provided that a fair solution were found as between ship and ship, the shelter deck type would suffer no injustice, while all shipowners would benefit by being able to get better designed ships at less cost owing to the elimination of useless and often harmful arrangements necessary to circumvent the tonnage laws, not to mention the fees for making the present elaborate measurements, which would become largely unnecessary.

There are many other ways in which the tonnage laws react unfairly between ship and ship, although perhaps not quite so striking or so ridiculous as these two. One concerns the measurement, or rather exemption from measurement, of double bottoms; others concern the way in which measurement is taken to the face of the frames of a ship and to the way in which sparring and ceiling are treated. In tankers, for instance, the framing is often made deeper than it need be simply to avoid the effects of tonnage measurement.

These are, in brief, some of the reasons which show why the present tonnage laws are unfair as between ship and ship, even if they were

admmistered and interpreted on the same uniform basis. But tonnage laws must inevitably be administered by a number of different authorities of different nationalities all over the world, and the more complicated they are, and the more definitions which lend themselves to differences in interpretation, the greater becomes the scope for further unfairness. The nation which applies a strict interpretation may well be handicapping its ships as compared with those of a nation where the interpretation is lax. If, however, some simple and easily checked tonnage laws were in operation, this inequality of treatment at least would be avoided.

The problem of finding a single and fair tonnage law is very similar to the problem which has faced those who have framed the rules by which racing yachts have been measured. In the case of racing yachts, the old tonnage rules were at first used, but as designers found ways of cheating the rules, new rules had to be produced. At one time, length between perpendiculars was one of the main ingredients of a formula, which was killed by the famous *Jullanar* in 1874, where the sternpost was placed very far forward. But the main object of all yacht rules is to provide a rule which, while giving a designer as much latitude as possible, tends to penalize all yachts which do not approach to the type which has been proved by experience to be the most suitable for the purpose. The problem for ships is easier than that for racing yachts, because there are considerations other than speed which tend to keep ships to the most suitable type. In fact, most ships would certainly have been better designed, and more efficient, if the tonnage laws had not existed when they were built.

It would, perhaps, be possible to amend the existing laws so as to remove the handicaps to design which they now present, but with each amendment the laws and their interpretation would almost inevitably become more and more complicated. But why preserve an unnecessary and laborious process for achieving a comparatively simple end? The main object of tonnage laws is to provide a yardstick by which to assess the dues which each ship shall pay for the use of ports and canals, and surely the simpler the law is, and the more easily checked it is, the better.

Some of those who have studied the matter have advocated displacement as the best yardstick on the grounds that displacement has already had to be measured for load line assignment. There is logic in this solution, but it is open to the argument that it would unfairly affect the ship with the full coefficient as compared with the ship with the fine coefficient. Moreover, it is not an easy matter to check the displacement of a foreign ship coming into your ports. Would not the fairest and simplest tonnage law be a formula composed of the three main dimensions which measure roughly the cost of building and maintaining the harbor or canal facillties—length, breadth, and draught of water? A formula based on these three is entirely logical for its main purpose—the assessment for dues; and, moreover, these three dimensions are so easily measured that there should be no room for differences in interpretation in different parts of the world.

The only difficulty—and not a very great one at that—in making up such a formula is the definition of length, breadth and draught. If we defined length as the overall length, the tendency would be for ships to have vertical stems and sterns which, apart from being unsightly, are not the most serviceable forms. If we define length as length between perpendiculars, the tendency would be for sternposts to be placed further forward than would be desirable. If we define length as length on the water line at full load draught, the tendency would be for excessive overhangs above water. It would appear, therefore, that whatever definition of length is adopted must be qualified in some way. A definition such as this might meet the case—"Length shall be the length on load water line, provided that the overall length shall not exceed the L.W.L. length by more than 2 per cent at the fore end, and 3 per cent at the stern. If the overall length

exceeds the L.W.L. by more than these amounts each excess shall be added to the length for tonnage purposes." Or, alternatively, length might be defined as "length between perpendiculars with an allowance of say 2 per cent forward and 4 per cent aft." This would bring us close to the definition used by the classification societies, which is roughly the length, on the water line, from the stem to the sternpost, with allowance of 4 per cent. This definition would probably need to be qualified by allowance of say 2 per cent at the fore end, and the allowance of the after end should not be confined to water line length but made to cover overall length.

Breadth presents less difficulty. The extreme beam, as presently measured by the Board of Trade, would give a fair measurement. The only qualification needed would be to allow the overhang of one or more unenclosed decks by say 2 per cent of the beam on each side, to meet the case of passenger vessels, or of docking platforms overhanging from the navigating bridge. Draught would be defined as the salt water draught to the Summer Load Water Line.

It is suggested that a formula

$$L \times B \times d/a \text{ coefficient} = \text{Tonnage}$$

would give a simple, fair, and easily measured tonnage for all ships, and would have the minimum hampering effect on design. Another alternative would be to use depth in place of draught. Some naval architects already use a formula Length×Breadth×Depth×a coefficient to estimate approximately the tonnage.

Admittedly, any change in the tonnage laws would involve a large amount of office work and issue of new tonnage documents for existing ships as well as an adjustment of the scales of charges to all ports and canals; but as a change in the present complicated and certainly unsatisfactory law is more than overdue, surely this opportunity should be taken to put the tonnage law on a simple basis. There is no need whatever to have two tonnages such as gross and net, and while we can never get rid of displacement tonnage and deadweight tonnage, it is suggested that if there is one universal tonnage on which the dues were paid in all ports and canals, this and this only would tend to be used for all statistics. So far as harbor and canal authorities are concerned, their main concern is the amount of revenue which they collect in dues. If therefore a change in tonnage laws alters the total amount of tonnage using the port or canal compared with the present position, the collecting authority has merely to apply a factor to its existing scale of charges. This factor would be easily found by taking an average selection of the ships concerned and comparing their tonnages under the old and the new conditions.

The minds of the laymen, the public, and the politician are continually being confused by the production of figures and statistics making use sometimes of one tonnage measurement, sometimes of another. No layman can easily understand the difference between gross and net tonnages. One simple universal tonnage law would make all shipping questions very much easier to understand and to handle. For the good of shipping as a whole, it seems obvious that the question must be tackled, and tackled in a radical way—tinkering with it will only waste time and energy for years to come and will perpetuate an anachronism, a fetish, to the services of which many good surveyors all over the world have given their life's work quite unnecessarily.

When I come to review the whole position, I think that I am entitled to put it even more strongly and to say that the present tonnage laws are a disgrace to the shipping community of the world and an insult to their intelligence. Their net result is:

(1) To waste the life work of a large number of skilled surveyors and much time in shipyard drawing offices, where men have to learn the refine-

ments of the tonnage laws and to become experts in the art of circumventing them by legitimate expedients;

(2) To produce several forms of tonnage figures confusing to the layman, and even to those in the industry;

(3) To encourage owners to make openings in some types of ship, thereby making them less seaworthy;

(4) To force designers to make machinery spaces much larger than they should be, leaving large empty spaces in the middle of the ship and discouraging new types of machinery;

(5) To give considerable latitude to their interpretation, allowing the lax to score over the strict;

(6) To make the tonnage of a loaded ship almost impossible to check; and, finally,

(7) To give anything but a fair assessment between ship and ship.

These seven evils, inherent in the present law, make it imperative that a clean sweep should be made, and a completely new and simple law evolved. No tinkering with the old law can be satisfactory. Any new, simple, well thought out and fair law would be of enormous advantage to the shipowners, the shipbuilders, and the engineers. And now is the time for it.

The root of the whole trouble has been the principle on which the old tonnage laws were based, that is to say, the earning capacity of the ship. If the alternative principle, which is obviously the right and equitable principle to use, namely, the cost of the service rendered to the ship, had been employed, the tonnage laws would not have got into anything like the muddle in which they are today. Even if they were to be revised now, and if they were to continue to be based on the wrong principle, they would almost certainly get into a muddle again. I cannot too strongly emphasize that, in my view, it is this fundamental mistake of principle which has caused all the trouble and which will continue to cause trouble until it is altered.

ELECTRONIC TUBES—HOW THEY WORK.—The electron is both the basic unit of electricity and a very real thing. Its mass and its charge, though minute indeed, were accurately measured by scientists many years ago. However, until creation of the electron tube, or vacuum tube, electrons were not free to be moved about as desired and their interesting and useful properties could not be studied and made use of. Four unique characteristics of the electron tube are responsible for its value in many applications. These characteristics are:

(*a*) Almost complete independence of electric frequency,

(*b*) Ability to control electric currents smoothly,

(*c*) Ability to control the movement and velocity of the speeding electrons by merely changing the electric potential of one of the electrodes inside the tube, and

(*d*) Ability to pass current only in one direction or, as it is often expressed, to act as a rectifier.

The comments above are based on a recent GE Science Forum talk by W. C. White. The following article is reprinted from the November, 1943, issue of General Electric Review.

In recent years, and especially during the war, electronic tubes have won such a place in industry that they are now performing countless tasks more efficiently and accurately than they have ever been done before.

For example, the high-frequency radio waves used to heat-treat steel for gears and tools are produced by electronic tubes; similar methods are used for bonding layers of plywood in the manufacture of plywood airplane propellers.

Electronic tubes are used to control accurately the speed of electric motors under varying load conditions; to provide stepless control of lighting circuits; to operate d-c motors from a-c lines; to convert alternating current into the vast quantities of direct current required for chemical plants, shipyards, steel mills, airplane factories, and other industries; and to operate automatic door-openers, burglar alarms, and a host of other devices.

There are electronic tubes so sensitive that they can measure the minute quantities of electricity in the muscles of the human heart; others are sturdy enough to carry 10,000 amperes for such operations as resistance welding.

Highly specialized instruments, such as the recording spectrophotometer, the electron microscope, electronic strain gages, and many others, have been developed around the electronic tube.

The tubes in home radio receivers and the picture tubes in television receivers are electronic, as are those used in the transmitters at the broadcasting stations. X-ray tubes used by doctors and dentists and in industry for detecting flaws in metal are electronic. Even the fluorescent lights in factories, stores, and homes are electronic tubes.

These are only a few of the uses to which electronic tubes have been put. A complete listing of only the industrial uses of these tubes would run into hundreds of applications. In almost every type of industry, they have been used to measure and control such chemical and physical quantities as acidity, color, temperature, speed, pressure, and time.

And engineers are daily finding new ways in which to put electronic tubes to work. Obviously, then, it is important to understand the operating principles of these tubes. The purpose of this article is to present these fundamentals in a simple and easy to understand manner.

Since there are many types of gas-filled and high-vacuum tubes, it is impossible to describe them all in an article of this size. However, the basic principles of all tubes are essentially the same, and a thorough understanding of one tube will make it easy to understand the operation of all electronic tubes. This article will describe the basic parts, construction, and operation of a specific tube. For this purpose, the widely used gas-filled FG-57 thyratron has been selected.

Inasmuch as there are two main classes of tubes, the high-vacuum and the gas-filled, an explanation of the basic differences between them is also given.

All the circuit diagrams in this article show the direction in which electrons flow, i.e., toward the *most positively charged potential.*

Tubes Control Current Flow.

Just as the flow of water in a pipe is the combined movement of a great many drops of water, the flow of electricity in a conductor, such as a copper wire, is the combined movement of a stream of electrons. Each electron carries a tiny quantity of electricity.

As long as electrons remain imprisoned in wire, control over them is limited. But if they can be liberated from the wire under controllable conditions—such as exist within a sealed glass or metal bulb—their usefulness can be greatly increased. And that is where the electronic tube fits into the picture.

When you connect an electronic tube into an electric circuit, you actually break the conductor, leaving a gap within a sealed enclosure. Electricity

must flow across this gap in order to close the circuit and permit current to flow again. An invisible stream of electrons—each electron carrying a tiny quantity of electricity—carries the current through the space in the tube.

During the split second when the electric current is a flow of "free" electrons in the tube, it is subject to a method of control which is not possible while electrons are imprisoned in the wire. The current can be stopped, started again, increased or decreased—instantaneously, smoothly, accurately, and without mechanical movement, noise, or vibration.

Basic Tube Parts.

Any electronic tube has four basic parts: (1) the cathode—a surface which gives off electrons; (2) the anode—a surface which receives electrons; (3) a glass or metal envelope; (4) terminals for connecting the tube into the circuit. Some tubes have a fifth part, the grid, which controls the current passing through the tube. Many tubes have a separate heating coil to bring the cathode to the temperature required to make it emit electrons. The FG-57 thyratron tube has all six of these parts.

In the following description of these basic tube parts, all letters A, B, C, etc., refer to the chart.

The Heater (A).

In most tubes the cathode is heated to render it emissive. There are two methods of heating the cathode, each of which has special advantages for certain applications.

(1) One method is to build the cathode in the form of a filament which is heated by its own resistance when a current is passed through it. These self-heated filamentary cathodes look much like the filament of an incandescent lamp.

(2) The other method is to use a separate heating coil inside heat conserving shields. This heating coil is called the heater.

The heater (A) is made of a metal with a high melting point, such as tungsten. Operating temperatures vary; in the FG-57 thyratron tube the heater operates at a temperature of about 1800 F and heats the cathode to about 1600 F.

The cathode is not heated to its operating temperature by the main current flowing through the tube. A separate circuit, fed by a low-voltage transformer or battery, supplies the current for heating either the filamentary-type cathode or the heater types.

The FG-57 thyratron tube requires 4.5 amperes at 5 volts in the heater circuit.

The Cathode (B).

When an electronic tube is connected into a circuit, the conductor is actually broken. The cathode inside the tube is one end of the broken conductor and the anode is the other. To re-establish a flow of current through the circuit, electrons must flow across the gap between the cathode and the anode.

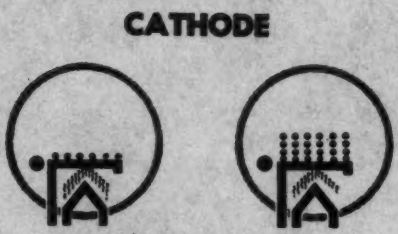
Electrons cannot get across this gap without assistance. They must first be released from the cathode so the anode can "pull" them across the gap. As mentioned, heating the cathode is one way to liberate electrons, and it is the method used in most tubes. When the cathode is heated electrons boil off its surface—much as steam is boiled off the surfec of water (B).

Heated cathodes—whether of the directly or indirectly heated type—are made of a metal having a high melting point, such as nickel alloy, coated with certain chemical compounds to increase its ability to give off electrons.

Whether a cathode should be directly or indirectly heated depends on the electrical characteristics desired for the tube and the materials used.

ELECTRONIC TUBES . . . how they w

A The heater in an electronic tube heats the cathode.



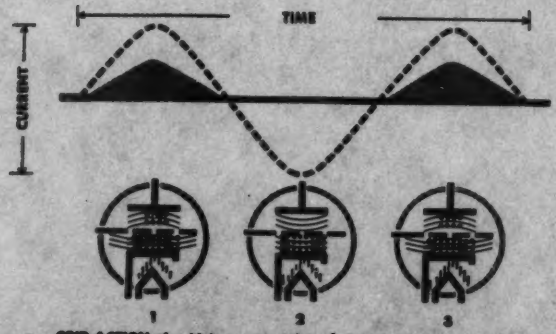
B When the cathode is heated, electrons boil off its surface—in much the same way as steam is boiled off the surface of water

C 1. When it is pos attracts electr 2. When it is n anode cannot attract e

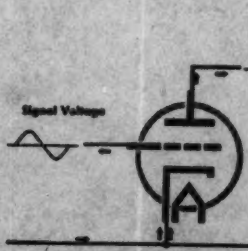
E Symbol for a gas-filled tube

F Symbol for a high-vacuum tube

G IONIZATION 1. An electron (negative charge) is knocked gas molecule by another electron from the cathode. 2. Th is now positive. 3. Positive molecules are called ions. neutralize most of the space charge (negative electric field cause passing from cathode to anode), permitting more electrons to re Ionization is responsible for the greater current-carrying capaci tubes

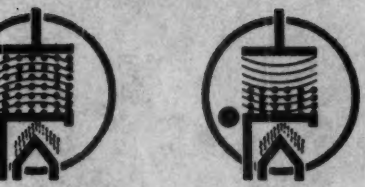


I GRID ACTION of a high-vacuum tube. 1. Grid is cancelling part of effect of anode; only part of electrons are reaching anode. 2. Anode is negative and does not attract electrons (no current flows). 3. Cycle repeats. Note cessations and uniform control of current flowing through the tube (solid area)



J AMPLIFIER—The plate-current wave take ly the same shape as the signal voltage. plate current is many times larger than current, this results in amplification

ANODE



1. When it is positively charged, the anode attracts electrons from the cathode. 2. When it is negatively charged, the cannot attract electrons

GRID

D 1. When more than a certain value of negative voltage is applied to the grid, electrons cannot reach the anode. 2. When the voltage on the grid becomes less negative, the grid suddenly permits the full flow of electrons to the anode. 3. Flow of electrons stops when anode voltage becomes negative. Cycle then repeats

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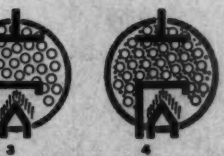

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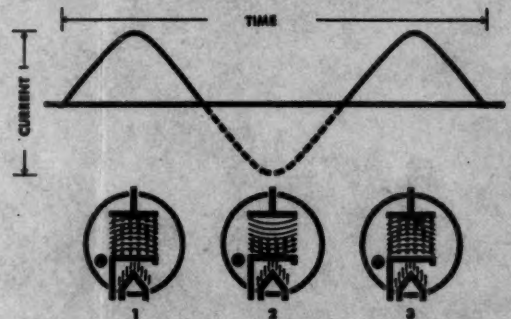
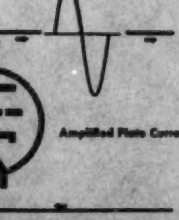


H RECTIFIER 1. Anode is positive and attracts electrons. 2. Anode is negative and does not attract electrons. 3. Cycle repeats—each step lasts 1/120 second. Note that negative portion of each cycle (dotted line) is eliminated



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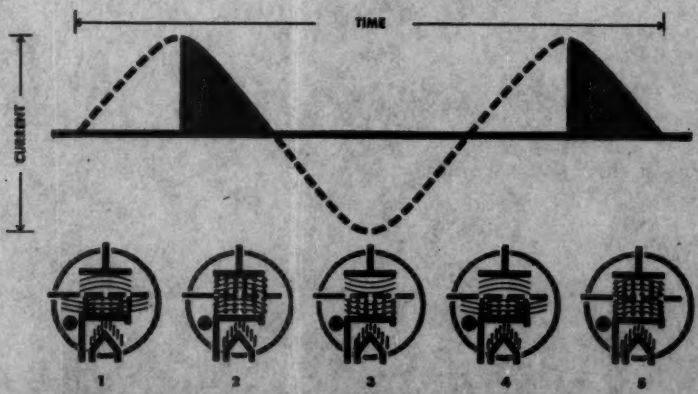

K GRID ACTION of a gas-filled tube. 1. Grid prevents flow of electrons—completely cancels effect of anode. 2. When less than a minimum voltage is applied to grid, full flow of electrons is suddenly released (solid area). 3. Anode is negative and does not attract electrons. 4-5. Cycle repeats

Filamentary (directly heated) cathodes heat more quickly than indirectly heated cathodes and are used where quick heating is desired. A much larger emitting surface can be heated efficiently with the indirect method. Therefore, indirectly heated cathodes are generally used in tubes which are expected to carry large currents.

Other methods of liberating electrons are used in certain types of tubes. Phototubes utilize light beams to liberate electrons from the cathode; glow tubes are actuated by the potential gradient (difference in voltage) between the cathode and the anode; mercury-pool tubes release electrons by means of an arc drawn between the surface of the mercury and the anode, which in some cases is started by a special electrode called the ignitor.

The Anode (C).

The anode is the other end of the broken conductor in the tube. It attracts the billions of electrons that are "boiled" off the surface of the cathode, thus establishing a flow of current through the tube (C). The anode is frequently referred to as the plate, and the current which flows through the tube is called the plate current.

While the anode (or plate) can attract electrons, it is incapable of giving off electrons as long as the tube is operating at the proper temperature.

When a positive voltage is applied to the anode, it attracts the electrons given off by the cathode. The reason for this is that electrons are always negatively charged, and opposite electrical charges attract each other. Therefore, if the tube is being used in a direct-current circuit, the anode must always be connected to the positive side of the circuit.

When a tube is connected into an alternating-current circuit, the anode is positive during half of each cycle. During this half cycle electrons leave the cathode and are attracted by the anode, resulting in a flow of current through the tube. During the half cycle when the anode is negative, electrons cannot reach the anode. During this half cycle, current cannot flow through the tube.

It is this characteristic of an electronic tube which enables it to act as a rectifier, changing alternating current into direct current.

The higher the positive voltage applied to the anode, the greater its ability to attract electrons, and the greater the rush of electrons to it—up to the limit of the ability of the cathode to give off electrons.

Different materials are used in fabricating anodes. They include tungsten, molybdenum, nichrome, graphite, nickel, and tantalum. Different shapes, also, are used. Material used and shape selected depend on the desired electrical characteristics and the amount of cooling necessary. (The anode must not be permitted to overheat.) The FG-57 thyratron tube has a nichrome anode.

The Grid (D).

The grid is the controlling electrode of the tube. It is located between the cathode and the anode in the path of the electrons. Some tubes have several grids to permit greater range of control over the plate or anode current.

The grid (D) in any gas-filled tube controls only the point in the cycle at which electrons will start to flow to the anode. For example, if the tube is being used to control a resistance-welding circuit in which the welding time is required to last only five one-thousandths of a second, the grid can be made to hold the current off until the last five one-thousandths of a second of the positive side of the cycle.

The grid in the FG-57 will not permit electrons to reach the anode as long as a certain negative voltage value is maintained on the grid. When

this voltage becomes less negative, the grid works like a trigger, suddenly permitting the full flow of electrons to start toward the anode.

Once the flow of electrons has started, the grid in any gas-filled tube loses control over the flow until the next cycle; that is, until the anode has become negative and stopped the flow of electrons.

After the flow of electrons has stopped, the grid regains control, and once more will prevent the start of the flow until voltage on the grid again becomes less negative.

The effect of the grid in high-vacuum tubes is quite different, for it never loses control over the flow of electrons. Instead, the flow is at all times proportional to the voltage applied to the grid, and any change in grid voltage causes a proportional change in the amount of current flowing through the tube.

In both the gas-filled and high-vacuum tubes, the power required in the grid is very small. Therefore, a relatively small amount of power in the grid circuit of an electronic tube can precisely control a much larger amount of power in the plate circuit.

The Terminals.

The terminals on the tube provide for connection of the cathode and anode into the main circuit, and connection of the heater and grid to their respective supply circuits.

There are three separate circuits in an electronic tube with a grid, and for each additional grid there is one additional circuit. Tubes without a grid have only the heater and cathode-to-anode circuits; or, in the case of tubes with unheated cathodes (such as phototubes and glow tubes) only the cathode-to-anode circuit.

The Envelope.

The chief function of the envelope is to maintain the vacuum in the tube. In some tubes it also has an important function in dissipating the heat created in the tube so as to keep it at a proper working temperature.

It is necessary to enclose the metal parts (electrodes) of the tube in a vacuum. If the cathode were to be heated to its operating temperature in the presence of air, the oxygen in the air would cause the cathode to burn up. Even if air would not result in the destruction of the cathode, it would cause corrosion of the metal parts, or would cause the formation of a coating of oxides, thus reducing the efficiency of these parts.

The envelope maintains controlled gas-pressure conditions, ranging from a very high degree of vacuum to the low pressure of gas used in the FG-57 thyratron tube.

In making a gas-filled tube, the air is first pumped out until a high vacuum is created. Then, before the tube is sealed, a tiny quantity of argon, neon, mercury, or some similar substance is inserted. (However, the pressure is still well below that on the outside of the tube.) This gas or mercury is chemically inert, which means that it will not combine with any of the metals used in the tube.

Basic Tube Types.

As previously mentioned, there are two basic types of electronic tubes. Both types are necessary, since each possesses characteristics either lacked by the other or superior to those of the other. Generally speaking, gas-filled tubes (E) are necessary where large amounts of current must be controlled, as in resistance welding. High-vacuum tubes (F) are necessary in jobs calling for continuous control of relatively small currents.

The major distinctions between gas-filled and high-vacuum tubes are these:

(1) High-vacuum tubes work like a throttle in that they provide continuous and uniform control of the amount of current flowing through the tube.

(2) Gas-filled tubes work like a switch. They are capable of accurately and instantaneously closing a heavy-current circuit.

(3) Gas-filled tubes will carry heavy current much more efficiently than high-vacuum tubes of the same size. This heavy current is conducted with a low and relatively constant voltage drop.

(4) High-vacuum tubes can be used as distortionless amplifiers; gas-filled tubes cannot.

It has already been explained that a gas-filled tube also is a relatively high-vacuum tube, for the gas it contains is at an extremely low pressure—far below atmospheric pressure. The presence of this small amount of gas, however, is the key to the difference in operation of the gas-filled and high-vacuum tubes.

Space Charge and Ionization.

In both the gas-filled and high-vacuum tubes, electrons are given off by the cathode. These electrons are drawn to the anode by the attractive force of its positive charge (since opposite electrical charges attract each other).

The effect of the billions of electrons passing between the cathode and the anode is to create a strongly negative electric field (since electrons are always negative). This field is known as space charge.

This negative field cancels part of the positive field created by the anode, thus reducing the ability of the anode to attract electrons. This is known as space-charge effect and is the basic reason for the inability of high-vacuum tubes to carry as large a current for their size as gas-filled tubes.

When a small amount of gas is placed in the tube it fills the space in the tube with gas molecules which, by comparison to the tiny electron, are very large. A gas molecule consists of a nucleus, which is positively charged, and a number of electrons which revolve about it much like the planets of a miniature solar system. There are just enough electrons to balance the positive charge on the nucleus and make the molecule electrically neutral.

Electrons emitted by the cathode travel with tremendous velocity and when one of them collides with a gas molecule the impact knocks one or more electrons out of the molecule. These molecules, which have lost some of their electrons and are now positive, are called ions. The process is called ionization (G).

The electrons knocked out of the molecules join with the stream of electrons moving toward the anode. The ions remain in the tube for a relatively long time and neutralize most of the effect of the space charge. The effect of ionization is to lower the voltage drop across the tube. Thus, ionization is responsible for the greater current-carrying capacity of gas-filled tubes.

Tube Output Diagrams.

The distinctions between gas-filled and high-vacuum tubes can best be understood when presented in the output diagrams for each tube.

Assume that a typical current—the 60-cycle, 110-volt alternating current—is being used in the cathode-to-anode circuit of the tube.

Such a current changes direction 120 times a second—twice each cycle. What is shown in (H) takes place in 3/120 of a second.

Because electrons can leave the cathode but cannot leave the anode, electronic tubes permit current to flow in only one direction and only

during the positive portion of each cycle. The negative portion of each cycle (represented by the dotted lines) is eliminated by the tube.

In other words, the current is being continuously turned on and off. It is on for 1/120 second and off for 1/120 second. The output of the tube is, therefore, a pulsating direct current. The pulsations can be removed by filters.

Diagram (H) illustrates the output of a tube having no control grid. Since it is utilizing only half of the current wave (one side of the cycle only), it is known as a half-wave rectifier. If two tubes are connected so that the cathode of one is connected to the anode of the other, thus conducting both halves of the cycle, the output will be an alternating current which can be controlled cycle by cycle. This type of circuit is used for resistance welding.

Assume now that we replace this rectifier tube with a high-vacuum tube containing a grid. Assume also that the grid is set to reduce the flow of current through the tube by one-half. Diagram (I) shows the output wave of this tube. (Dotted line represents the maximum output of the tube; solid area is the output permitted by the grid.)

The diagram shows that the controlled-output wave has the same shape as the full-output wave. This means that the tube has exerted a continuous and uniform control of the current during the entire period of flow.

This continuous and exact control is the reason for using high-vacuum tubes for amplification. For example, in (J) if we had a very faint current—too weak to be used directly—it would be used as the signal voltage on the grid of the tube. Every variation in the shape of its wave, no matter how small or complex, would be faithfully reproduced in the output wave of the tube. This output wave would be an amplified (enlarged) "image" of the grid wave.

Now, suppose that the high-vacuum tube is replaced by a gas-filled tube in which the grid is set to reduce the flow of current through the tube by one-half. The output diagram for this is shown in (K).

In this case, the output diagram does not have the same shape as the uncontrolled output wave. Instead, it shows that the current did not flow at all during the first quarter of a cycle. At this point, the full flow of current suddenly came on and remained on for a quarter of a cycle. During the negative half cycle (the period when the anode is negative and prevents electrons from flowing) the current remained off, as it did in the vacuum tube.

Since the controlled wave does not have the same shape as the uncontrolled wave, the gas-filled tube cannot be used as an amplifier without distortion. Because the current is being switched on and off, part of the signal voltage would be lost during each cycle.

The diagrams illustrate the basic differences between high-vacuum and gas-filled tubes. The high-vacuum tube works like a rheostat, the gas-filled tube works like a switch.

Although the same average amount of current was permitted to flow through both tubes, the method of control was different.

In the gas-filled tube, the current was completely off for a quarter of a cycle and completely on for a quarter of a cycle—in short, the average flow was reduced by turning it off for part of the time.

The high-vacuum tube reduced the average flow of current by throttling down the flow—just as the flow of water from a faucet is reduced by turning the handle.

NEW SUPERCHARGER FOR MARINE ENGINES.–This paper was read at the War Engineering Production Meeting of the Society of Automotive Engineers. The authors are Alf Lysholm, Chief Engineer of the Swedish firm of Aktiebolaget Ljungstroms Angturbin; Ronald B. Smith, Manager of Research Engineering for the Elliott Company, and W. A. Wilson, Mechanical Division Engineer of the Elliott Company. The paper is reprinted from The British Motor Ship, October, 1943.

Our national emergency has served to accelerate the application of supercharging to internal-combustion engines. While the extreme in the utilization of auxiliary charging is, by virtue of necessity, associated with the gasoline aircraft engine, the more moderate application of the scheme has resulted in higher specific output for all types of engines. To the Diesel engine designer, supercharging is particularly significant, since it has not yet been possible to achieve as effective use of the air charge in the compression-ignition engine as in the spark-ignition type.

Of the several schemes for supercharging now in practice, the mechanically driven positive displacement compressor has much to recommend it, particularly on small high-speed engines, where simplicity and cost are paramount. For Diesel-engine applications where high torque is a necessity at reduced speed the displacement compressor is effective inasmuch as the engine behaves as an air absorber with about the same speed delivery characteristics as the compressor, with the result that a full air charge can very nearly by maintained at all operating speeds.

There are a number of incentives for the development of a more efficient supercharging compressor. Until now the centrifugal machine, with an adiabatic efficiency of from 70 to 75 per cent, has held a margin in this respect. While the efficiency of the mechanically driven supercharger bears a rather obvious relation to the friction loss of the engine, it has an equally important influence on the charge delivered to the cylinder. An engine with a displacement of D and a compression ratio ϵ, supercharged to a pressure ratio of ρ and unscavenged will absorb per stroke an air charge of

$$\frac{D\eta_v}{V_1}\ \rho\left[1+\frac{1}{\epsilon-1}\left(1-\frac{1}{\rho}\right)\right]\left[1+\frac{\frac{1}{\Delta T}}{T_1}\right]\text{lb}$$

where ηv is the volumetric efficiency of the normally aspirated engine, V_1 and T_1 are the specific volume and the absolute temperature of the atmospheric air, and ΔT is the temperature rise in the compressor. In the derivation of this relation it is assumed that the heat transferred to the inducted charge by the manifold and the cylinder walls results in a constant temperature increase irrespective of the charging conditions. The temperature rise in the compressor proper is inversely proportional to its efficiency, with the result that a smaller charge is inducted and a higher compression temperature is attained in an engine operating with an inefficient compressor. For a four-cycle engine charged to 6 to 7 pounds per square inch an increase in the adiabatic blower efficiency from 60 to 70 per cent increases the output by 1.2 per cent through a direct reduction in power absorption, and increases the inducted charge by 2.2 per cent. With the engine maintaining a constant ratio of excess air the combined yield is slightly over 3 per cent in excess power. Under part-load operating conditions the effect is an even larger percentage. A more efficient compressor becomes increasingly advantageous when cylinder scavenging is introduced inasmuch as the scavenging air is supplied as a direct power

loss. Until scavenging is fully adopted by the four-cycle Diesel engine, including alterations in the valve timing and exhaust-manifold system following the Büchi principles, the full advantages of supercharging, particularly as they influence the effectiveness of air utilization, are not realized. For the moment much thought is being devoted to pressure charging in excess of 10 pounds per square inch in conjunction with auxiliary starting means, and under these conditions the advantages of high compressor efficiency are even more obvious.

At present, the most common form of positive displacement supercharger is the Roots-type blower in which the air is trapped between the lobes and carried around to the exit where a back rush of air from the discharge chamber raises the pressure of the charge. The result is essentially a constant-volume pressure rise and a spare-type indicator card. In the Centric and the Sulzer machines an attempt has been made to overcome the disadvantage of the Roots type of compressor by providing, in a rotary machine, means for adiabatically compressing the charge by reducing its volume before the discharge ports are uncovered. These machines incorporate rubbing or sliding surfaces which are internally lubricated—a condition which is intolerable on the induction system of an engine. The advantage of compression within the cylinder is evident from the work ratio for an ideal adiabatic compression to that for constant volume compression

$$\frac{W_{AD}}{W_{v=Const}} = \frac{k}{k-1} \frac{\frac{k-1}{k}}{\frac{\rho-1}{\rho-1}}$$

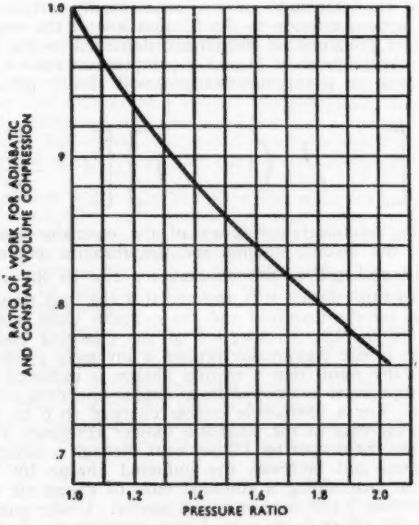

Fig. 1.—Comparison of adiabatic and constant volume compression.

where K is the ratio of specific heats and ρ the pressure ratio of compression. This function is a comparative measure of the internal efficiency of the two operating cycles and is plotted in Figure 1. With the 7 to 8 pounds per square inch charging pressure commonly employed there is a 14 per cent inherent advantage to the adiabatic cycle and with the 15 to 20 pounds per square inch pressures now being contemplated, a 25 per cent gain is a potentiality.

The Elliott-Lysholm rotary compressor was developed in an attempt to overcome the obvious shortcomings of vane and reciprocating-type compressors by combining the advantages of high rotating speed, simplicity, freedom from rubbing contact and internal lubrication, inherent in the Roots machine, with the desirable characteristic achieved by internal compression. The development was undertaken in Sweden in 1934 and is covered by a number of foreign and domestic patents. Approximately 30 machines have now been built and tested, varying in size from 15 c.f.m. to 10,000 c.f.m.

In this compressor the air is transported diagonally by virtue of the pair of helically lobed rotors which interact to provide an axial as well as a cross seal. The distinguishing feature of the compressor lies in its action on each charge of suction air after it is sealed off from the inlet and before it is brought into communication with the discharge. A charge is initially enclosed in the space bounded by the tooth flanks, casing bore and end walls. The rotor helices are so chosen that a particular thread space is completely filled and sealed off from the suction just as it is entered at its opposite end by a co-acting lobe on the other rotor. Further rotation establishes an axial seal which separates this charge from the charge in the

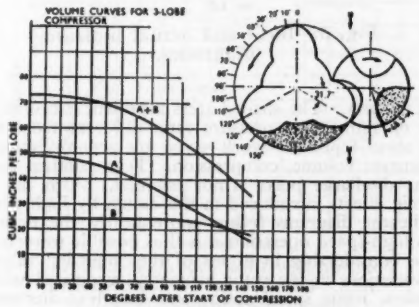

Fig. 2.—Compression relationships for Lysholm rotors.

succeeding grooves. As rotation proceeds, this seal moves axially, effecting a reduction in volume of the charge and a substantially adiabatic compression. When the leading lobes of the grooves pass the boundaries of the discharge port, the compressed medium is brought into communication with the discharge. The location of the port thus determines the "built-in" compression ratio. A typical relationship of charge volume to rotation is illustrated in Figure 2.

Theoretical indicator cards, assuming no leakage and no dynamic effects, are shown in Figure 3 for an Elliott-Lysholm compressor operating against

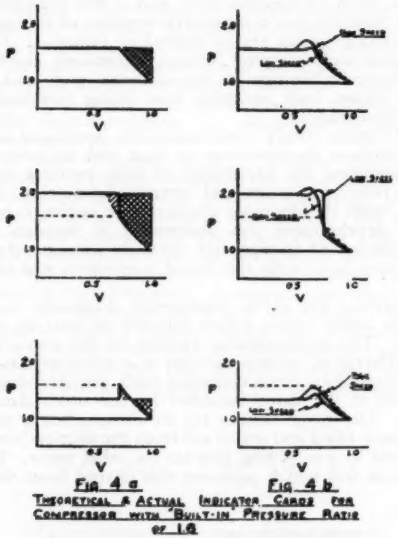

Fig 4 a

Fig 4 b

Theoretical & Actual Indicator Cards for Compressor with Built-in Pressure Ratio of 1.6

Fig. 3.—Ideal and actual indicator diagrams.

various back pressures. The shaded areas represent excess work performed as a result of operating at a pressure ratio different from that "built-in." The hatched areas represent work saved by accomplishing adiabatic, as opposed to constant volume, compression. Tests confirm that an accurate location of the discharge ports is not essential. With the slight leakage and the dynamic effects introduced in the design by high peripheral speed, the actual indicator diagrams follow more nearly the type illustrated in Figure 3.With high-speed operation it is thus possible to secure an optimum operating ratio considerably in excess of the "built-in" value.

Numerous variations in the rotor geometry, associated with lobe height, number of lobes, rotor speed ratios and length-to-diameter ratio, have been incorporated in the machines already constructed to suit particular applications. Good space utilization is attained with a few lobes having a large height, whereas better port areas are available when a larger number of lobes are employed. In general, tooth ratios from 1 : 1 to 1 : 1.5 have been found most effective.

The tooth profiles are virtually all addendum on the rotor with convex lobes, while the mating rotor has a tooth form largely dedendum, except for a small addendum and dedendum modification on the respective rotors to replace otherwise sharp corners by radii and fillets. The convex rotor absorbs approximately 110 per cent of the power, the concave rotor being driven by the air. The small torque transmitted by the timing gears maintains the teeth in continuous contact on one side. For certain positions in the rotation of the rotors, small pockets are known to exist at each end which do not communicate with their respective ports. This

condition is often relieved by appropriate grooves in the rotors, but on small compressors this refinement is generally unnecessary.

An Elliott-Lysholm machine for supercharger applications having a capacity of from 400 to 500 c.f.m. is illustrated. To eliminate the disadvantages associated with their distortion and high temperature expansion, light metals have not been employed. Completely equipped and constructed in cast iron, the compressor weighs only 80 pounds—a feature which can be attributed to the effective use of high rotating speed inherent in the Lysholm arrangement. The compressor is driven through a coupling or by means of a belt at from 2000 to 3000 Rpm. Speed-increasing gears, transmitting the driving torque to the timing gears, are incorporated within the casing and are self-lubricated from the oil reservoirs at each end. Hardened and ground airplane-type gears are employed. Leakage of oil into the intake of the machine is eliminated by a carbon nose ring rubbing on the lapped faces of the timing gears, whereas on the outlet end air leakage from within is throttled through close-fitting carbon rings. The rotating parts are supported on anti-friction bearings designed for an average life of 50,000 to 100,000 hours. A complete mock-up of the rotating system was operated continuously in the laboratory for nearly 1000 hours at full speed and overload torque before a final design was attempted. The exit porting was arranged for a built-in compression pressure ratio of 1.3.

The adiabatic and volumetric efficiencies of the compressor, measured with a calibrated orifice and with the machine cradled in a dynamometer, are shown in Figure 4. A maximum efficiency of 72 per cent has been achieved, with performance over much of the speed range in excess of

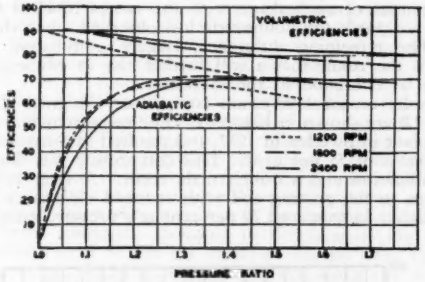

Fig. 4.—Adiabatic and volumetric efficiencies for a 400-500 c.f.m. Elliott-Lysholm compressor.

70 per cent for pressure ratios as high as 1.7. The flatness of the efficiency characteristic with respect to speed and pressure ratio has been exhibited by all Lysholm compressors. It is in part due to the fact that the decrease in volumetric loss with speed is compensated by the increased dynamic losses at the inlet and outlet; and in part to the increase in mechanical efficiency as the leakage losses increase with pressure ratio. A flat performance characteristic is advantageous inasmuch as it means that although the compressor is designed for a fixed compression ratio, its behavior under other than design conditions is so flexible that any mechanical scheme for adjusting the discharge porting is unnecessary. In Figure 5 is shown the temperature efficiency of the machine as determined from the temperature rise measured with calibrated N.A.C.A. probes. A maximum efficiency

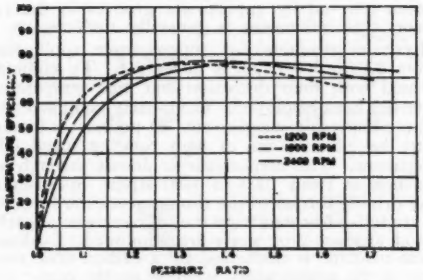

Fig. 5.—Thermal efficiency of a 400-500 c.f.m. compressor.

value of 76 per cent has been determined in this manner—the difference between the adiabatic and temperature efficiency being largely the result of 1 Hp. mechanical loss associated with the bearings, gears and lubricating means.

The efficiency attained by this compressor has not been achieved by closing the clearances in the casing and rotor. The design specified 0.005-0.006 inch clearance which, under actual conditions, was slightly exceeded. Static measurements of air leakage with the rotors blocked in the casing indicate only two-thirds the volumetric loss observed when the compressor is rotating. The remaining difference, largely a dynamic loss, can be controlled with the result that a still further rise in efficiency is possible without resort to reduction in the clearance.

The test results on an 1100-c.f.m. compressor designed for a pressure ratio of 1.5 to 2.0 are shown in Figure 6. The measurements were made at the De Laval plant in Sweden in 1937, and resulted in a maximum adiabatic efficiency in excess of 75 per cent. This compressor was designed with a 4-6 tooth combination and a 6-inch male rotor. A later test made after slight alterations to the porting and with reduced clearances resulted in a maximum efficiency in excess of 79 per cent at a pressure ratio of 2.0.

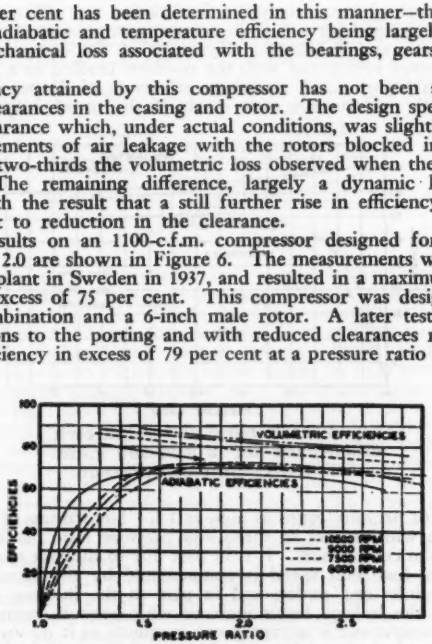

Fig. 6.—Adiabatic and volumetric efficiencies of 1,100 c.f.m. compressor.

The Lysholm type of compressor has already been built in sizes large enough to supercharge the biggest Diesel engines that have been designed. For engines operating at constant speed, simple by-pass control can be readily incorporated in the design for use at light load, to reduce the friction drag on the engine. For the high-pressure supercharging now contemplated the compressor offers an efficiency exceeding that attained even by the single-stage centrifugal type and the complication of auxiliary gear transmissions. Two-stage units operating at discharge pressures as high as 100 pounds per square inch have also been built. Light-weight units for aircraft service are also feasible, and are in fact in the experimental stage of operation abroad.

BARGE PLACES 90-TON BREAKWATER BLOCKS.—This is an abstract of C. H. Cruttwell's recent article in Engineering News Record. It is reprinted from the Current Topics section, Journal, Franklin Institute, September, 1943.

A dumping barge of unusual design and construction was used to transport and place 90-ton concrete blocks in a breakwater recently completed at the port of Leixoes, Portugal. The job involved extending an existing breakwater about 3300 feet, of which the first 1300 feet was a coursed wall built of heavy rectangular concrete blocks, while the remaining 2000 feet, subject to considerable wave action, was a submerged-mound type of granite rubble base capped by the 90-ton concrete blocks. It was for construction of this outer end of the breakwater that the special barge was required. The special feature of the barge, which was 141 feet long overall, with a 32-foot overhang at the bow, permitted a close approach to the breakwater when dumping the blocks. Designed as a heavy beam, the central portion of the barge supported a track of two pairs of railroad rails carrying a tipping car. This beam and track arrangement was at the bottom of a trough extending nearly the full length of the hull. A seven-ton steam-powered winch in the stern of the barge propelled the tipping car through four 1¼-inch steel cables. Two water-ballast tanks, placed aft, could be filled or emptied as required to control the inclination of the deck under different loadings. Mounted in the bow of the barge were five cast steel rollers over which the concrete blocks were tipped. The 90-ton cubical concrete blocks were loaded in line along the center of the barge, resting on hardwood battens bolted longitudinally along each side of the trough. Seven blocks, 12 feet 3½ inches × 10 feet 8 inches × 10 feet 10 inches, constituted a barge load. After loading, the barge was towed to the breakwater site and made fast to six buoys by 7-inch coir rope cables and the car was hauled forward causing the block to roll over the bow rollers and, once the center of gravity had passed the equilibrium point, the block tipped over.

BOOK REVIEWS.

QUESTIONS AND ANSWERS FOR MARINE ENGINEERS. From Marine Engineering and Shipping Review, Boilers and Engines. Compiled by Captain H. C. Dinger, U.S.N. (Retired). Published by Simmons-Boardman Publishing Corporation, 30 Church St., New York, N. Y. Price $2.00.

This Booklet is a second edition of Books I and II of the series, combined into one volume. Other volumes of this group have been reviewed from time to time. The present volume presents in question and answer form material of value to operating engineers on the subjects of Boilers and Engines. The compilation by an experienced and practical engineer can be of value for improving practical operation or for preparation for license examinations.

OTHER BOOKS RECEIVED.

AMERICAN PADDLE STEAMBOATS, by Carl D. Lane. Published by Coward-McCann, Inc., 3 West 45th St., New York 19, N. Y. 224 Pages. Price $6.00.

ASSOCIATION NOTES.

The Newcomer Medal for Achievement in Steam has been awarded to Rear Admiral H. G. Bowen, U. S. Navy, a Past President of the Society, with the following citation:

> "In consideration of his long record of service with a group of Engineers, whose work is now such a vital part of the life of our Nation, and particularly in view of his outstanding advocacy of the advance in Steam Engineering in this branch of our National Defense, as represented by the increase in steam pressures and temperatures used in equipment installed in its ships"

Officers for 1944.

The following have been elected officers of the Society for the calendar year 1944:

President:
Rear Admiral C. L. Brand, U. S. Navy.

Secretary-Treasurer:
Captain J. E. Hamilton, U. S. Navy.

Member of Council:
Mr. Bryce W. Blair.
Mr. T. H. Bossert.
Captain E. E. Brady, U. S. Navy.
Captain R. W. Bruner, U. S. Navy.
Captain J. B. Dow, U. S. Navy.
Commander R. B. Lank, U. S. Coast Guard.
Captain Paul F. Lee, U. S. Navy.
Mr. J. B. Woodward.

The amendment proposed to the By-Laws providing that an officer of the Naval Reserve shall be a member of the Council was adopted by vote of the Society. No nomination of a

Reserve officer could be made pending action on the proposed amendment. Adoption of the amendment, therefore, created a vacancy in the Council. Lieutenant Commander Walter B. Armstrong, U.S.N.R., was appointed by the Council to fill the vacancy.

Membership.

The following have joined the Society since the publication of the November, 1943, issue of the Journal:

Naval.

Armstrong, David M., Lieutenant, U. S. N. R.

Armstrong, Richard W., Ensign, U. S. N. R., B. O. Q. No. 1, Lewis Park, Norfolk, Va.

Ayer, William H., Lieutenant, U. S. N.

Beardslee, F. Pardee, Lieutenant, U. S. N. R., 2923 Macomb St., N. W., Washington, D. C.

Bebb, William M., Midshipman, M. N. R., Brounwood, Ga.

Boys, William G., Lieutenant, U. S. N. R., care Supervisor of Shipbuilding, U. S. N., Brown Shipbuilding Co., Houston, Texas.

Bradford, Louis M., Lieutenant, U. S. N. R., care Supervisor of Shipbuilding, U. S. N., Tampa, Florida.

Brown, Jacque L., Lieutenant, U. S. N. R., 1425 Rhode Island Ave., N. W., Apt. 304, Washington, D. C.

Browne, Oscar M., Jr., Navy Yard, New York, N. Y.

Bullard, B. B., Lieutenant, U. S. N.

Burke, Edmund F., Lieutenant, U. S. C. G., 45 Christopher St., New York, N. Y.

Burroughs, John, Lieutenant, U. S. C. G. R., 3523 Martha Custis Drive, Parkfairfax, Alexandria, Va.

Bush, Edgar D., Lieutenant, U. S. N. R., Room 1-W-100, Navy Building, Washington, D. C.

Chiswell, W. B., Commander, U. S. C. G., Coast Guard Receiving Unit, Alameda, Calif.

Clark, Sumner C., Ensign, U. S. N. R.

Cleare, K. S., Lieutenant, U. S. N. R., Mail, 3716 San Luis St., Tampa, Fla.

Crawford, John William, Lieutenant, U. S. N.

Davis, Harry E., Lieutenant Commander, U. S. C. G., Naval Engineering Division, U. S. Coast Guard Headquarters, Washington, D. C.

Furlong, Donald, Lieutenant, U. S. N., 403 Memorial Drive, Cambridge 39, Mass.

Garay, Paul N., Ensign, U. S. N. R.

Gordon, Floyd, Machinist, U. S. N.

Gould, William K., Lieutenant, U. S. N. R., Office of Supervisor of Shipbuilding, San Pedro, Calif., Mail, 440 Coronada Ave., Long Beach 4, Calif.

Green, William P., Lieutenant, U. S. N. R., care Hugh Green, M. W. Douglas Ave., Jacksonville, Ill.

Greenleaf, William C., Lieutenant, U. S. N. R.

Hankins, B. M., Ensign, U. S. N. R., Apt. B, 510 Linden Ave., Long Beach 2, Calif.

Hedgecock, R. H., Lieutenant, U. S. N., care Senior Naval Office, Mass. Inst. of Technology, Cambridge, Mass.

Henderson, Douglas B., Lieutenant Commander, U. S. C. G., 2412 Manhattan Ave., Manhattan Beach, Calif.

Herbert, Clarence, Lieutenant Commander, U. S. C. G., U. S. C. G. District Office, Alaska Building, Seattle, Wash.

Higgins, George C., Lieutenant Commander, U. S. N. R., 6429 Aloerton Ave., Forest Hills, N. Y.

Jakomas, Daniel V., Lieutenant, U. S. N. R.

Kelly, Douglas V., Ensign, U. S. N. R., 1 Canyon Road, Berkeley, Calif.

Keune, Lyle J., Lieutenant, U. S. N.

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